GENERAL MOTORS

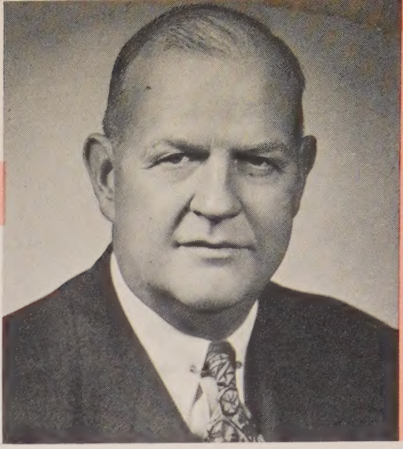
ENGINEERING JOURNAL

Volume 5 October-November-December 1958 Number 4

for educators in the fields of engineering and allied sciences

INDEX ISSUE

AN OUTLINE IN YARN STARTS THE DESIGN



On the Challenge of the Space Age

It has been the practice of General Motors to hold an Engineering and Science Educator Conference each year. The 1958 Educator Conference, attended by 28 faculty members from colleges and universities throughout this country and Canada, was recently concluded.

The purpose of the Educator Conference is twofold—first, to acquaint educators with GM product, production, and research activities, and second, to provide the educators with the opportunity to discuss matters which they feel to be of particular interest and importance to our joint responsibility of educating and using technical personnel to the maximum advantage of all.

During the year which elapsed between the 1957 and 1958 Educator Conferences, the dramatic launching of the first Sputnik occurred. This event, more than any other single event in recent history, has created a greater self-searching evaluation of the effectiveness of the American educational system and the increasing need by industry for scientifically trained personnel.

The history of mankind from the beginning has been divided and characterized by certain *Ages*. The present and the future for years to come has been and will be designated as the *Space Age*. During the past year much discussion has taken place and much has been written regarding whether or not U.S. elementary, secondary, and higher educational institutions measure up to the requirements of a highly competitive and extremely scientific Space Age.

While our Conference this year did not deal specifically with education in the Space Age as such, nevertheless the demands of the Space Age, particularly in the fields of engineering and science education, were very much in the background and affected to a substantial degree the subjects selected by the educators for discussion with us. It was obvious from their discussions that in approaching the educational requirements of the future no revolutionary ideas should be proposed. One of our educator guests stated that education's ultimate objective, to meet most effectively the requirements of the Space Age of today and the future, should be to teach a student to think, to make decisions, and to learn how to act upon the decisions to accomplish the job at hand. Three things would be needed to achieve this objective: first, *intellect*—the power of knowing, the capacity for higher forms of knowledge, the ability to imagine, and the ability to think and to understand; second, *skill*—the ability to use one's knowledge or intellect effectively; and third, *tools*—the instruments to accomplish one's intellectual tasks, instruments such as books, laboratories, computers, in short, anything that helps to get the job done quickly and effectively with the least expenditure of precious man hours.

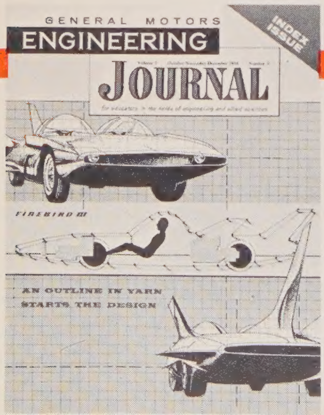
Today, all industry is alert to the necessity of more effective utilization of educational institutions and more effective use of their product. All industry, with the help of educational institutions, has undertaken definite programs to meet the increasing needs of the Space Age.

For example, it has been the practice of General Motors to employ high school science and mathematics teachers and students of technical colleges and universities during the summer months. Such a program has contributed substantially to the experience and capabilities of the participants. Also, General Motors encourages its engineers and scientists to join and actively participate in technical societies. In addition, various GM Divisions and Staffs provide the opportunity for their engineers and scientists to continue their education after graduation and employment.

Our conclusions after this very useful period of association and consultation with our recent Educator Conference guests may be summarized as follows: this Space Age in which we now live is in every sense an Age of the engineer and scientist. The need and opportunity for engineers and scientists in this country's defense security and in its day-to-day and future progress represents a challenge and opportunity never before equalled in all the past Ages of mankind.

C. R. Osborn

C. R. Osborn,
Vice President and
Group Executive



THE COVER

The automotive stylist's threefold objective is: satisfy the imagination; provide for comfort, visibility, and control; and provide for the placement of basic mechanical components. In styling Firebird III—the latest GM experimental gas-turbine powered vehicle—a new approach was used to achieve these objectives, as illustrated in this issue's cover designed by artist Ernest W. Scanes. Stylists used colored knitting yarn and pins to wrap the exterior lines of the body around cut-outs of such basic components as the front mounted accessory-drive engine and the rear mounted gas turbine engine. This allowed the shape of the body to be changed by merely shifting a pin. When the exterior shape was agreed upon, the design was then photographed and served as a model for blueprints.

CONTENTS

TECHNICAL COMMENTARIES

Page

The Use of an Electrical Model to Analyze Torsional Vibration in a Diesel Engine, *by Allen B. Hamilton, Electro-Motive Division* 2

The Application of Dielectric Heat to Body Interior Trim Fabrication, *by Ralph M. Stallard and Paton M. Zimmerman, Fisher Body Division* 7

Some Applications of Stress Analysis Techniques in Improving Casting Designs, *by Louis H. Ravitch, Central Foundry Division* 22

The Filing and Maintenance of Patents in Foreign Countries, *by Fred E. Jones, Patent Section, Detroit Office* 38

GM LABORATORIES AT WORK

The Application of a Magnus Effect Flow Meter as a Laboratory Teaching Device, *by Wesley J. Trathen, General Motors Institute* 12

Firebird III: A New Concept in Automotive Engineering 33

ENGINEERING PROGRESS AND FACILITIES

Simple Two-Wheel Trailer Measures Road Surface Coefficient of Friction, *by Paul C. Skeels, General Motors Proving Ground* 14

Dynamic Air Gaging Reduces Cycle Time in Precision Measurement, *by Alex H. Joyce and Kenneth E. Ruff, General Motors Process Development Staff* 28

ENGINEERING NEWS

New Gyroscope Developed as an Aid in Teaching in the Field of Physics and Allied Sciences 21

Notes About Inventions and Inventors, *Contributed by Patent Section, Detroit Office* 39

Technical Presentations by GM Engineers 51

OTHER FEATURES

A Typical General Motors Institute Laboratory Problem: Develop an Alignment Chart to Find Air-Fuel Ratios from an Orsat Analysis for the Combustion of C_8H_{18}, *Faculty Members in Charge: Donald R. Jenkins and Frank N. Crall; G.M.I. Cooperative Student: Charles Copeland, AC Spark Plug Division* 46

Solution to the Previous General Motors Institute Classroom Problem: Determine the Static and Total Temperatures of a High Temperature, High Velocity Gas Stream, *Faculty Member in Charge: Kenneth W. Woodfield; G.M.I. Cooperative Student: Rodger Bloomfield, AC Spark Plug Division* 47

Solution to the Previous Problem: Determine the Effect of Changes in Ambient Temperature on an Automotive Air Suspension System, *by Eric Dietrich, Buick Motor Division, and Leonard B. Wocholski, General Motors Institute* 54

Contributors to the October-November-December 1958 issue of GENERAL MOTORS ENGINEERING JOURNAL 57

Index to Volume 5 GENERAL MOTORS ENGINEERING JOURNAL 60

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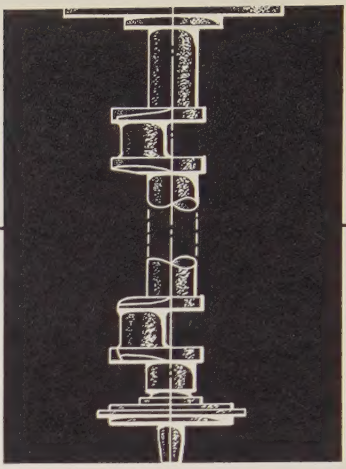
The Use of an Electrical Model to Analyze Torsional Vibration in a Diesel Engine



Torsional vibration analysis of a Diesel engine requires the expenditure of considerable effort and time, especially when a purely analytical approach is used. To reduce the complexity and labor involved with such an analysis, Electro-Motive Division engineers apply the electro-mechanical analogy concept, which allows a mechanical system to be thought of in terms of electrical circuit theory. In its application, the torsional system is first idealized as a network of concentrated inertias, pure torsional springs, and pure viscous damping elements. An electrical model of the torsional system is then arranged in accordance with the schematic arrangement of the mechanical system. Electrical constants also are established according to the analogy used. Known currents are then fed into the electrical network with an audio oscillator connected successively to points where harmonic torques are known to be applied in the corresponding mechanical system. Varying the input frequency allows several resonant frequencies to be located in the system. Voltages are then measured at as many points desired to obtain data for subsequent calculations of vibration response and mode shape characteristics. The voltages obtained may be added vectorially with known engine torque configurations to provide displacement or stress data at various points in the torsional system. The use of the electrical model permits torsional vibration data to be obtained without the need for running a full scale test engine.

Analysis of the torsional vibration characteristics of multi-cylinder Diesel engines is generally a time-consuming activity at best. The vibration analysis associated with engine design is likely to compound the consumption of time because trial and error solutions generally must be sought in systems where changes in one or two parts can have profound effects on other parts in the system. The application of electro-mechanical analogy and the use of electrical models in vibration analysis, however, permits designers to achieve a sound arrangement of parts in a reasonable period of time.

The electro-mechanical analogy theory[1] shows that a valid electrical model composed of inductors, capacitors, and resistors can be studied in lieu of mechanical networks composed of springs, masses (inertias), and friction elements (dampers). Voltages and currents measured on the electrical circuit can then be converted into vibratory velocities and torques.

To illustrate this procedure, the method used by Electro-Motive engineers to obtain the torsional vibration response of a six-crank Diesel engine will be discussed. The problem confronting the engineer consisted of determining the natural frequencies, the relative displacement of the various inertias in the system, and the torques and stresses in the neighborhood of the critical speeds. (A critical speed is one where the torque impulses produced by the combustion process come "in tune" at a natural frequency of the spring-mass system.)

Analogous Electrical Circuit Based on Mass-Elastic Diagram

The crankshaft of the six-crank Diesel engine was idealized in the conventional manner into a system of flywheels (inertias) connected by massless shafts (torsional springs, Fig. 1). From this, the analogous electrical circuit immediately took shape (Fig. 2). The electrical circuit included the necessary measuring and excitation apparatus, such as a cathode ray oscillograph or vacuum tube voltmeter and a variable frequency audio oscillator.

Scale factors were developed in terms of the electro-mechanical analogy (Table I). These factors allowed data to be transformed from the mechanical system to the electrical system. In addition to the scale factors, electrical constants also were established, based on a given set of mechanical constants (Table II).

Torsional Vibration Determined by Measuring Voltages

After the electrical model was assembled, measurements were made. Natural frequencies of vibration were determined by connecting the oscillator and voltmeter at a convenient location in the network, adjusting the oscillator output to some convenient current level, and then searching through the available frequency range for resonant conditions (maximum voltages accompanied with minimum input currents). For the six-crank Diesel engine, the first three natural frequencies were found at 595, 1,030, and 2,330 cycles per second.

Amplitudes may be measured, of course, at any point in the electrical model and full advantage is taken of this opportunity when studying engine vibration problems.

The torsional vibration response was determined electrically by measuring voltages with a voltmeter (an oscilloscope also could have been used) at a point in the electrical circuit equivalent to a point on the Diesel engine which was easily accessible to torsional vibration measuring equipment. For the six-crank Diesel engine under discussion, point *F* (Fig. 2) was used because of its accessibility on the actual engine and because a comparison was desired between the electrical and actual mechanical systems. The oscillator was tuned to the natural frequency of the mode of vibration being investigated—for example, 595 cps—and then connected successively to points corresponding to those in the engine

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[*] For Mr. Hamilton's biography and photograph, please see p. 63 of the April-May-June 1958 General Motors Engineering Journal. This is the second paper contributed by Mr. Hamilton, assistant chief structural engineer, Electro-Motive Division.

By ALLEN B. HAMILTON*
Electro-Motive
Division

The practical application of electro-mechanical analogy

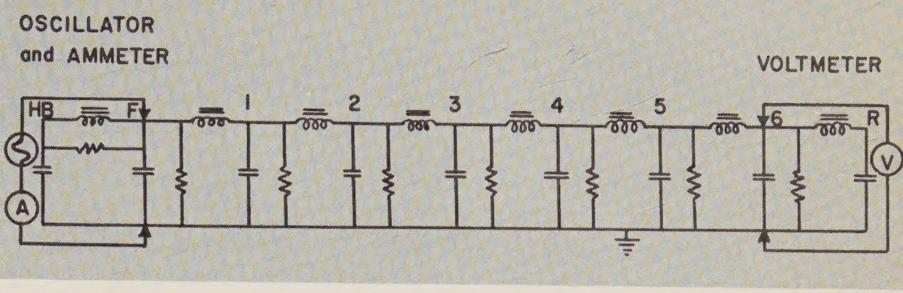

Fig. 2—The mass-elastic system of Fig. 1b, including the harmonic balancer *HB*, the front details *F* (accessory gear train), the cranks 1 through 6, and the rear details *R* (timing gear train), can be converted by analogy to the electrical equivalent circuit shown here. Added to the circuit are the necessary measuring and excitation apparatus, which includes an oscillator, ammeter, and voltmeter. Voltages and currents measured on the circuit are then converted into vibratory velocities and torques.

where input torques were received (points 1 through 6, Fig. 2).

The voltages measured at point *F* were recorded for each input station, with the input current held constant at 1.0 milliamperes. (A current amplitude of 1.0 milliampere was taken to represent a torque of 10,000 in-lb.) An alternative method would have been to apply the oscillator input at point *F* and then read voltages at this and other points in the system to determine complete mode shape and vibration response data. Either of these two methods would produce identical results. The validity for the interchangeability of these two methods rests on the Theorem of Reciprocity[2].

After all electrical measurements for the modes of vibration were made and the results tabulated, conventional elastic curves were drawn (Fig. 3).

Electrical Model Provides Complete Engine Vibration Data

Up to this point the electrical model accomplished as much as the conventional Holzer tabulation method used to obtain the natural frequency of a shaft system. In addition, however, the voltages measured on the electrical model gave a measure of the mechanical vibratory amplitudes. The method by which this was done can be explained by means of the following sample calculation for one of the harmonics in the first mode for the six-crank Diesel engine.

The harmonic orders which fell in or near the engine operating speed range were investigated for each mode of vibration. The various critical engine speeds were then determined by the following relationship:

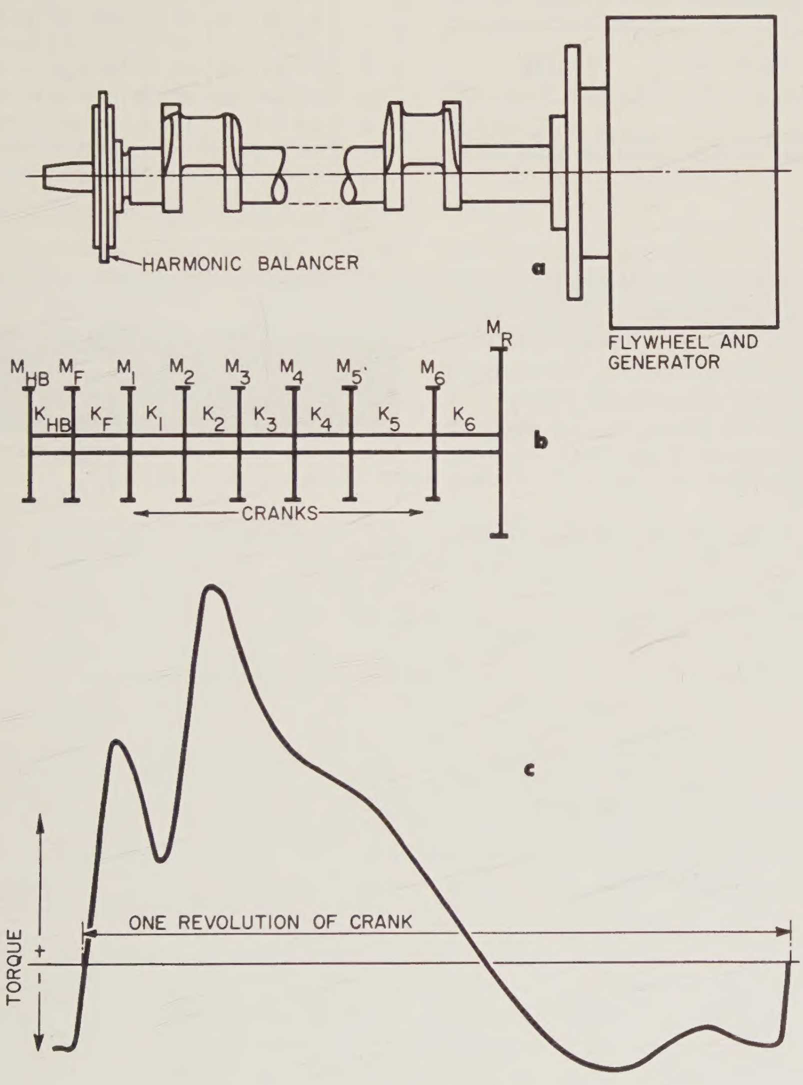

Fig. 1—A primary step in the development of an electrical model to obtain torsional vibration characteristics of a multi-cylinder Diesel engine is to change the crankshaft system (a) into a mass-elastic system (b) consisting of flywheels (inertias *M*) connected by massless shafts (torsional springs *K*). Torsional damping is applied at appropriate locations and an externally applied vibratory torque appears at each crank (c), which is representative of the variable explosion torques.

MECHANICAL TO ELECTRICAL SCALE FACTORS

MECHANICAL	ELECTRICAL	CONVERSION FACTORS
INERTIA M (in-lb sec^2)	CAPACITY C (microfarad)	$C = (0.000556)(M)$
SPRING K (in-lb/radian)	INDUCTANCE L (henry)	$L = \frac{1}{K}(18)(10^6)$
DAMPING D (in-lb sec/radian)	RESISTANCE r (ohm)	$r = \frac{1}{D}(180)(10^6)$
FREQUENCY ω (radian/sec)	FREQUENCY Ω (radian/sec)	$\Omega = 10\,\omega$
TORQUE F (in-lb)	CURRENT i (ampere)	$i = F(10^{-7})$
VELOCITY v (radian/sec)	VOLTAGE e (volt)	$e = v(18)$

Table I—To solve mechanical problems by electrical means requires the development of scale factors, such as those shown here, which allow data to be transformed from the mechanical system to the electrical system.

$$R_n = \frac{(60)\,(f_m)}{n}$$

where

R_n = critical engine speed for the nth harmonic order (rpm)

f_m = resonant frequency for mechanical system (cps)

n = harmonic order number.

To obtain the vibratory displacements at point F, the following expressions were used, based on the scale factors of Table I:

$$v = \frac{e}{18}$$

$$\Theta = \frac{v}{\omega} = \left[\frac{e}{(18)\,(2\pi)\,(f_m)}\right]\left[\frac{180}{\pi}\right]$$

where

Θ = displacements (degrees)

f_m = mechanical frequency (cps).

As a practical matter it was convenient to measure a-c electrical quantities with meters calibrated in root mean square (rms) units, while torsiograph data were read conveniently in double amplitudes and the torque input values were based on single amplitudes. The conversion from rms electrical quantities into peak-to-peak torsional displacements was made according to the following relationship:

$$\Theta = \Theta'\,(2\sqrt{2}) = 1.43\left(\frac{e}{f_m}\right), \text{ degrees}$$

double amplitude where e is measured in rms volts.

The value of Θ for each harmonic was determined as follows in accordance with the vector configurations for the engine. This was done in a manner consistent with the concept that the input response to the input torques (and currents) at each crank of the engine is proportional to the ordinate of the elastic curve[3] at such a point:

$$\Theta = \frac{(1.43)\,(e_f)\,(\Sigma_a)}{f_m}\,(F_n)\,(10^{-4})$$

where

Θ = torsional displacement at point F (degrees double amplitude per 10^4 in-lb torque input)

e_f = voltage measured at point F

Σ_a = vector sum of ordinates to elastic curve for points of torque input

f_m = mechanical natural frequency (cps)

F_n = harmonic torque amplitude per crank (in-lb).

The vector sum of ordinates Σ_a was based on vector configurations for the six-crank Diesel engine, as summarized in the following table:

Harmonic Order	Crank Vector Angles (Degrees)					
	1	2	3	4	5	6
1	0	120	240	240 + 26	120 + 26	0 + 26
2	0	240	120	120 + 52	240 + 52	0 + 52
3	0	0	0	0 + 78	0 + 78	0 + 78
4	0	120	240	240 + 104	120 + 104	0 + 104
5	0	240	120	120 + 130	240 + 130	0 + 130
6	0	0	0	0 + 156	0 + 156	0 + 156

The harmonic torque amplitude F_n for each harmonic order is summarized in the following table:

HARMONIC ORDER	HARMONIC TORQUE (in-lb)
1	67,000
2	18,000
3	9,000
4	0
5	9,800
6	9,400

MECHANICAL AND ELECTRICAL CONSTANTS

		MECHANICAL SYSTEM			ELECTRICAL SYSTEM		
		INERTIA (in-lb sec^2)	SPRING (radian per in-lb)(10^{-9})	DAMPING (in-lb sec per radian)	CAPACITY (microfarad)	INDUCTANCE (henry)	RESISTANCE (ohm)
	HB	33.7	115	3,600	0.0188	2.07	50,000
	F	32.2	5.42	180	0.0179	0.0975	10^6
CRANKS	1	29.8	6.66	180	0.0166	0.120	10^6
	2	23.9	6.66	180	0.0133	0.120	10^6
	3	23.9	6.66	180	0.0133	0.120	10^6
	4	23.9	6.66	180	0.0133	0.120	10^6
	5	23.9	6.66	180	0.0133	0.120	10^6
	6	29.8	7.16	180	0.0166	0.129	10^6
	R	5,000	—	—	2.78	—	—

Table II—Summarized here are the mechanical and electrical constants used in a vibration analysis of a six-crank Diesel engine.

To illustrate the method of calculation, the sixth order harmonic in the first mode of vibration for the six-crank Diesel engine is used as an example as follows:

$$f_m = \frac{595}{10} = 59.5 \text{ cps}$$

$$R_6 = \frac{(60)\ (59.5)}{6} = 595 \text{ rpm}$$

Σ_{a6}:

CRANK	a	VECTOR ANGLE
1	0.925	0°
2	0.800	0°
3	0.666	0°
4	0.494	156°
5	0.325	156°
6	0.150	156°

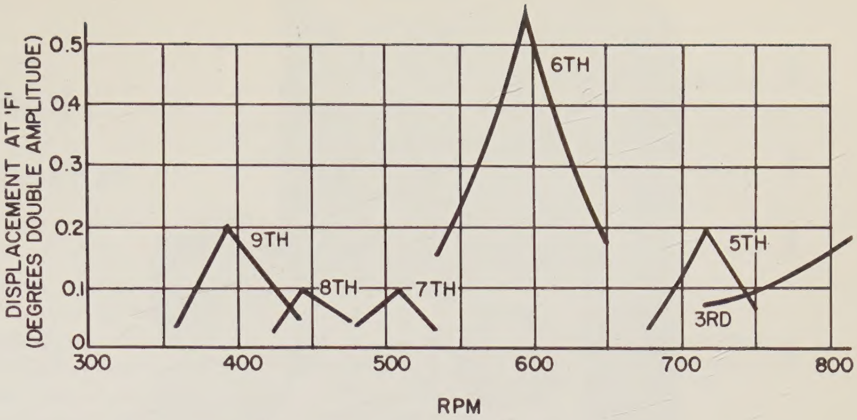

Fig. 4—To determine the overall torsional vibration response, amplitudes for all the harmonics must be calculated. Once the response is determined for a basic design, changes in design then can be evaluated quickly in terms of their resonant frequencies and comparisons of voltages against those obtained on the basic arrangement.

The above resultant was easily calculated by combining the sums of the front and rear halves of the engine by the Law of Cosines. The final results are as follows:

$\Sigma_{a6} = 1.55$

$F_6 = 9{,}400$ in-lb per crank

$e_f = 16$ volts rms

rms VOLTS AT POINTS TABULATED FOR INPUT
CURRENT OF 0.707 MILLIAMPERES rms AT "F"

MODE	HB	F	1	2	3	4	5	6	7	CPS
FIRST	33.5	16	14.8	12.8	10.5	7.9	5.2	2.4*	0.6	595
SECOND	7.8*	6	6.1	5.9	5.2	4.2	3.0	1.5*	0.12	1,030

*INDICATES A NODAL SHAFT—THAT IS, A SECTION OF SHAFT CONTAINING A POINT OF ZERO VIBRATORY DISPLACEMENT

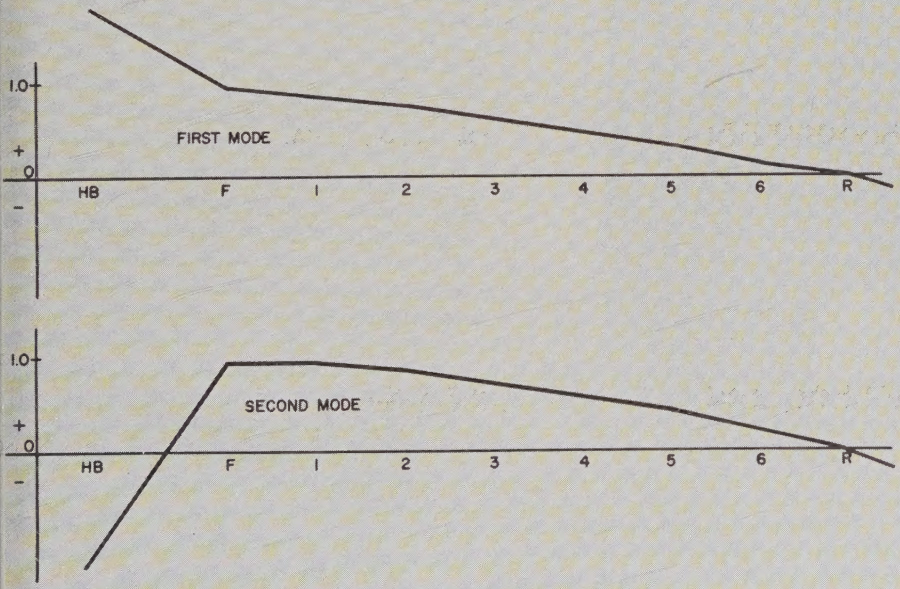

Fig. 3—Tabulated at the top are rms voltage values measured at point F (Fig. 2) for the first two modes of vibration. From this tabulation, the conventional elastic curves shown at the bottom for each of the two modes are drawn with the voltages for each mode referred to the voltage at point F (e_F) as unity.

Therefore,

$$\Theta_6 = \frac{(1.43)\ (16)\ (1.55)\ (9{,}400)\ (10^{-4})}{59.5}$$

$\Theta_6 = 0.56°$ double amplitude.

Knowing the value for Θ_6, the shape of the elastic curve, and the stiffness and diameters of the engine shafts, the torques and stresses due to this vibration were calculated. In the particular case being used as an example, the maximum vibratory torque T_6 occurred in the crank nearest the generator and had the following value:

$$T_6 = (\Theta_6)\ (K)\ (\Delta a)$$

$$T_6 = \left(\frac{0.56}{2}\right)\left(\frac{\pi}{180}\right)\left(\frac{1}{(7.16)(10^{-9})}\right)(0.19)$$

$$T_6 = \pm 130{,}000 \text{ in-lb.}$$

This torque, acting on a 6½-in. diameter crankpin, gave the following nominal torsional stress S_t amplitude:

$$S_t = T_6\left(\frac{16}{\pi d^3}\right) = \pm 2{,}400 \text{ psi.}$$

The amplitudes for the other harmonics were calculated in a similar manner so that the overall vibration response could be determined (Fig. 4). Having determined this response for a basic design it was then possible to make design changes and to evaluate them quickly in terms of their resonant frequencies and comparisons of voltages with those obtained on the basic arrangement. Thus, an experimental determination was possible on the electrical model which was equivalent to a test on a complete full scale engine. In the electrical model, the voltmeter served as a substitute for torsional vibration instruments used on the real engine.

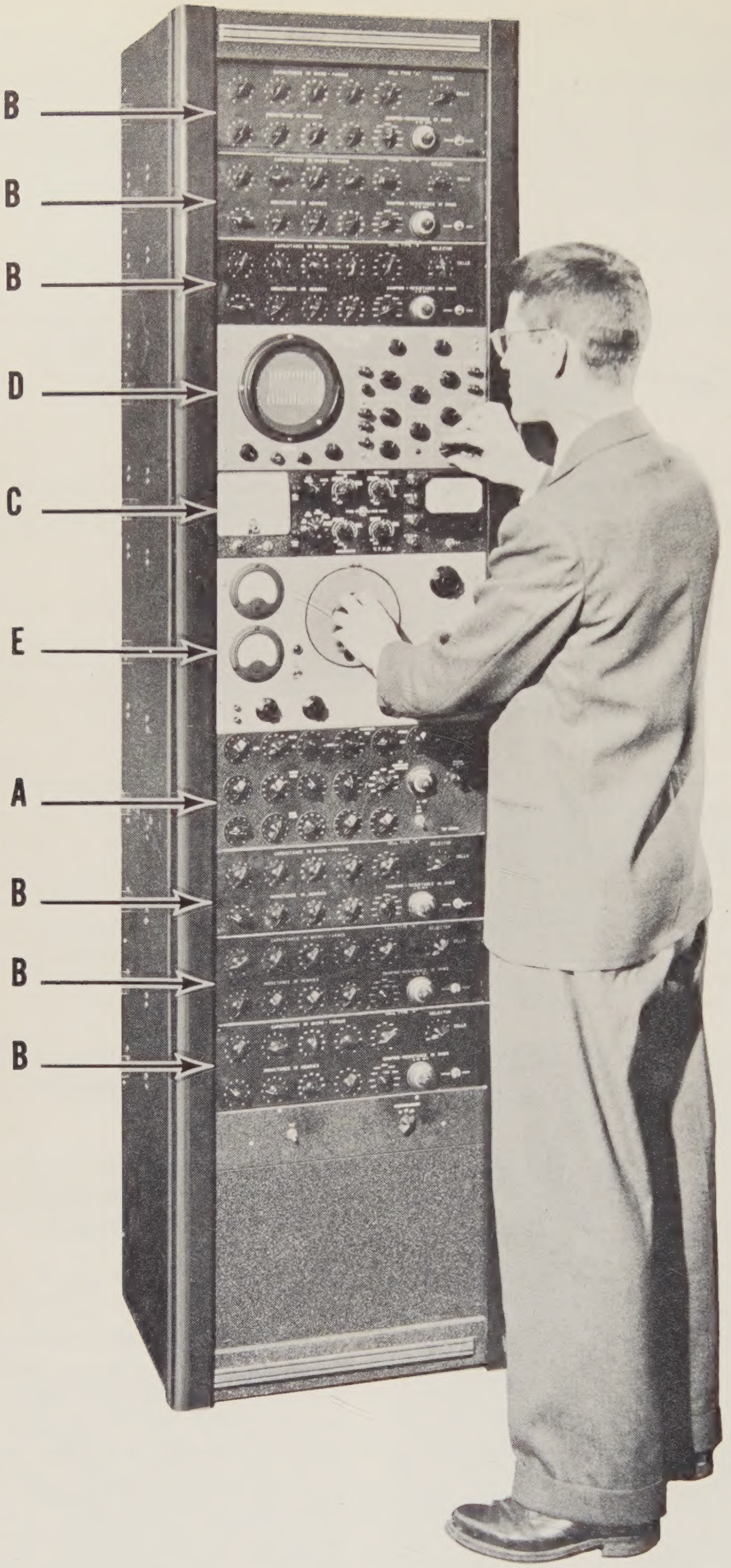

Fig. 5—A specially constructed panel specifically designed by Electro-Motive engineers to utilize the principles of electro-mechanical analogy allows one engine out of a line of 6, 8, 12, or 16 cylinder engines to be selected for torsional vibration study. The analogue equipment consists basically of an engine panel *A*, which houses the analogue components for 6, 8, 12, or 16 cylinder engines; six cell panels *B*, which may be used to represent additional spring-mass systems (the cell panels may be connected by means of switches to the ends of the engine or to one another); a control or "read-out" panel *C*, which permits connection of the various measuring devices and oscillators separately to any point in the engine panel or to the cells (the control panel also houses a vacuum tube voltmeter and ammeter); a cathode ray oscilloscope *D*; and an audio oscillator *E*.

Special Equipment Designed to Perform Vibration Studies

To take full advantage of the electro-mechanical analogy, Electro-Motive engineers constructed equipment (Fig. 5) specifically designed to make torsional calculations on a given line of Diesel engines. This restriction on the function of the equipment made it possible to build up the basic engine analogue in one electrical panel. By means of a simple switching arrangement one engine out of the line of 6, 8, 12, or 16 cylinder Diesel engines built by Electro-Motive can be selected for computation.

In operation, variable condensers are provided at the free and drive ends of an engine selected for study. Provision is made to attach the equivalent of a damped spring-mass set of variable proportions at the free end of the engine for the purpose of permitting studies of dynamic vibrational absorbers. Additional variable "damped spring-mass" sets are provided which can be connected by selector switches to others of the same kind or to either end of the engine under study. The addition of a built-in oscillator, oscilloscope, vacuum tube voltmeter, and a switch panel to make the necessary selections of instrument connections to the analogue model results in a fairly universal vibration calculation device.

Summary

The application of electro-mechanical analogy to the study of Diesel engine torsional vibration problems has greatly reduced the time usually involved with such work. Complete data on the vibration characteristics of an engine may now be obtained with an electrical model instead of running a test on a full scale engine. The information provided by the model allows design changes to be made and evaluated quickly with a subsequent saving in design time and fabrication expenditures.

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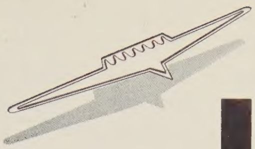
The Application of Dielectric Heat to Body Interior Trim Fabrication



By RALPH M. STALLARD
and PATON M. ZIMMERMAN
Fisher Body Division

New styling demands and new trim materials for automobile body interiors have made necessary new interior fabrication processes. The sewing techniques of the past have been replaced by embossing processes to join and decorate the trim materials. The two embossing methods used most are the hot platen and dielectric methods. The difference between these methods is the manner in which heat is generated for the embossing. The dielectric method induces heat within the trim material itself, while the hot platen method heats the trim with hot presses. The dielectric method is the more flexible of the two, and provides bodymakers with a versatile tool for fabricating complex trim arrangements.

Internally induced heat enables selective embossing of trim materials

MODERN automobiles owe much of their attractive appearance to the trim used to line the inside of the body and to upholster the seats. The public has indicated a preference for the complex decorative effects and multiple color schemes of modern trim design, and the present styling trend is placing greater emphasis on body trim.

Conventional sewing techniques used in the past for joining trim materials and for providing decorative effects were not capable of meeting these new styling demands economically. As a result, embossing processes using heat and pressure have been adapted to trim processing both for joining and decorating trim materials.

The two major kinds of embossing processes now used in the automobile industry for fabricating trim assemblies are hot platen embossing and dielectric embossing. Of the two processes, dielectric embossing is the more flexible and offers the greatest advantages. Dielectric embossing has opened up entirely new approaches to trim design and has become a valuable styling tool.

Dielectric Heating Induces Internal Heat

Dielectric heating is a selective heating process that develops heat internally in an electrical insulating material. Dielectric embossing has been confused with hot platen embossing, since both processes use heat and pressure, but they are different in the way the heat is generated.

Hot platen embossing uses heated embossing blades that "iron" impressions into the trim material. This heat also passes through the surface of the trim material to melt an adhesive that is used to bond together the layers of the trim sandwich, which consists of the visible cover trim, the padding, the adhesive, and the backing material (Fig. 1).

In comparison, dielectric embossing blades are not heated. Dielectric heating processes use an electric field to generate heat directly within the materials. Dielectric heating does not force external heat into the materials or pass a direct electric current in the normal sense through them. Instead, the sandwich of trim materials is put between two metal plates in a dielectric embossing press and an electric field is established between these plates. The electric field can be made to generate heat within the trim materials.

The energy for the electric field comes from an electric power source connected to the two plates of the dielectric embossing press. It is important to remember, however, that in dielectric heating, the electrons do not pass from one plate to the other through the trim sandwich. The trim materials are insulators and cannot normally conduct electrons.

Instead of current flow, the forces used in dielectric heating are caused by charges on the plates; one plate is positive and one is negative. These oppositely charged plates set up between themselves electrical forces of attraction and repulsion which are used to heat the trim materials.

The forces in the electric field can influence the movement of the trim molecules, since these molecules are not electrically symmetrical. The measure of the electrical asymmetry is called the dipole moment. This means that each molecule has built into it two poles with opposite charges, one positive and one negative.

Since opposites attract, the molecules

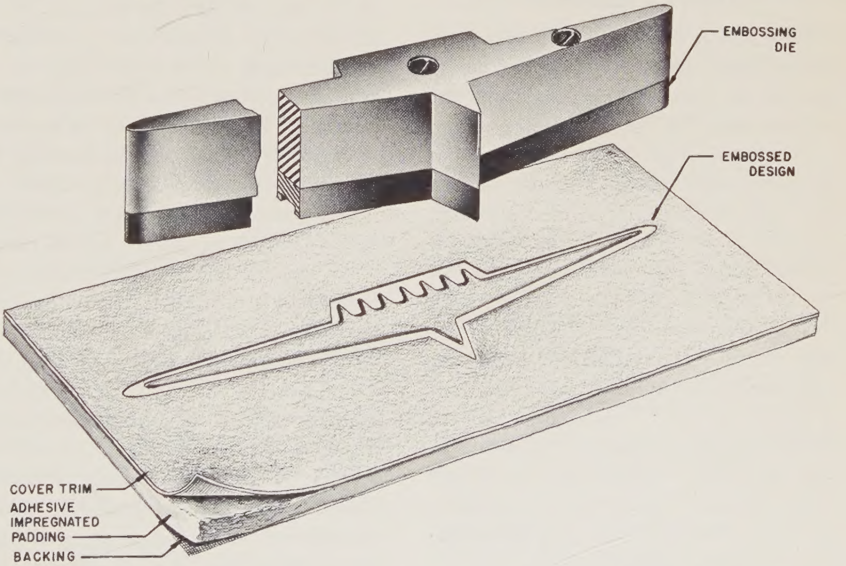


Fig. 1—The trim material visible in an automobile body interior is only the outside layer of a trim sandwich. A typical sandwich consists of the cover trim, which is the outside, visible layer, a second layer of adhesive impregnated padding, and a final layer of backing material. These three layers are bonded together during the embossing process, which also creates design patterns in the trim.

of the trim material line up with their negative poles toward the positive plate and their positive poles toward the negative plate (Fig. 2). Of course, just lining up between the plates does not cause the molecules to move enough to get very hot. But, in dielectric heating, the plates are connected to an alternating power supply so that the charge on the plates is repeatedly reversed.

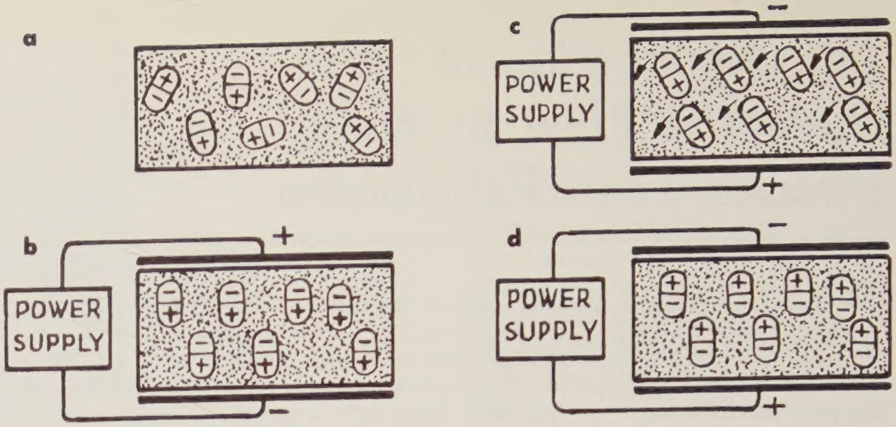


Fig. 2—The dipole molecules which are present in the trim material (a) orient themselves electrically when opposite electric charges are built up on surrounding plates (b). Although the heat generated within the material by this action is small, it can be increased to effective embossing temperatures by rapidly alternating the charges on the plates (c, d), thereby causing continuous agitation of the dipole molecules.

As the plate charges reverse, the dipole molecules rotate to keep the positive poles toward the negative plate and the negative poles toward the positive plate. This rotation of the molecules produces the heat needed within the trim materials for embossing.

It is important to remember that, practically speaking, work can be done by the electric field only if it changes polarity. Heat results from the work done by the *alternating* electric field.

The change from electric energy to heat energy that takes place in one cycle is so small that a very large number of cycles each second is required for the practical application of dielectric heating. In Fisher Body Division's dielectric heating equipment, the frequency is fourteen million cycles a second (14Mc) and the voltage is as much as 4,000 volts. The dielectric heating takes place in about 10 seconds, and the resulting molecular movement produces temperatures in the adhesive that are about 325°F, which is adequate for good bonding.

Three Major Units Comprise Dielectric Equipment

Dielectric embossing equipment consists of three major sections: the power section, the oscillator section, and the work section (Fig. 3).

The function of the power section is to convert low voltage, alternating current from the plant supply lines into high voltage, direct current. The oscillator section changes the high voltage, direct current from the power section into high voltage, high frequency, alternating current. In the work section, the high voltage, high frequency, alternating current is converted into heat energy in the material being embossed. The work section is located between the lower and upper platens of an air or hydraulically operated press used to supply the necessary embossing pressure.

An insulated electrode, or "cribbing," isolates the high voltage output of the oscillator from the rest of the press, so that a voltage gradient can be developed across the part being heated. Electrical shielding around the work section prevents the high frequency oscillator output from causing radio frequency interference during the embossing cycle.

The embossing blades that press the decorative lines into the heated trim material are not heated themselves, since they are metal and are unaffected by the electric field. They do absorb heat, however, from the trim materials. When the dielectric heating is stopped, the embossing blades quench and set the adhesive. This bonds the layers under the decorative lines where the embossing blades have pressed the layers together.

In dielectric heating, it is not necessary to heat the whole trim sandwich to the same temperature. This is what is meant by a selective heating process. By the proper choice of sandwich materials, the greatest heat can be generated in the adhesive so that it can flow through the padding. Only enough heat is needed in the trim to soften it for embossing, and the padding and backing materials can be chosen for their ability to generate only enough heat to give a good bond.

Another factor that sometimes contributes to heat in dielectric embossing is *ionic conductivity* (Fig. 4). In addition to having dipole molecules, some insulating materials also contain ions—charged atoms or molecules. Some ions are charged positively while others are charged negatively. When an insulating material is placed in an alternating electric field set up between two oppositely charged plates, the negative ions move toward the positive plate and the positive ions move toward the negative plate. Since the charge on the plates is rapidly reversing in dielectric heating, there is considerable back and forth movement of the ions, which creates heat through friction.

It should be emphasized that ionic conductivity is not electrical conductivity in the normal sense. Ions are larger than electrons and are more restrained in their motion than electrons, which move freely through a conductor when it is conducting electricity.

Ionic conductivity is of special importance in the dielectric heating of foam

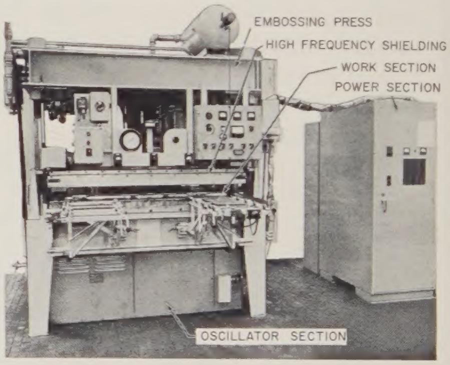


Fig. 3—The dielectric embossing equipment at Fisher Body Division consists of the work, power, and oscillator sections. The power section provides the oscillator section with high voltage, direct current. The oscillator section converts this current to high voltage, high frequency alternating current, which is converted into heat energy in the material being embossed by the work section. The work section is between the platens of an air-operated press. Safety features include a latch to prevent the upper platen from dropping if the air pressure fails, double palm buttons for the operator, and shielding which completely encloses the work section before the high voltage is applied. When two operators are stationed on opposite sides of the press, one cannot initiate operation unless the other's loading tray is in the back position, effectively preventing him from reaching the press. All electrical panels are interlocked to disconnect electrical voltages when any of the access doors are opened. Built-in safety blocks swing into position whenever the press requires die changing or maintenance.

rubber and felt riser materials which are normally poor in absorbing high frequency electrical energy. Before dielectric embossing, moisture is deliberately introduced into the foam rubber and felt which serves to increase the capability of the materials to absorb high frequency electrical energy by increasing their ion content.

Dielectric Constant and Power Factor Determine Heat Developed

Two important properties, the dielectric constant and the power factor, determine how much heat an insulating material will develop in an alternating electric field.

Dielectric Constant

The dielectric constant k of a material is the ratio between the parallel capacitance of a capacitor, or condenser, using that material as the dielectric and the capacitance of the same electrode configurations using vacuum as a dielectric. The dielectric constant gives an indication of the ability of the dipole molecules in a material to rotate when subjected to an electric field.

In trim embossing, the trim materials are stacked in a sandwich so that the dielectric constant becomes important for another reason. The dielectric constant determines, and is directly proportional to, the electrical capacity of an insulating material. Where several insulators are put in series with each other, as in the case of the trim, padding, adhesive, and foundation board in the trim sandwich, the insulator with the least electrical capacity will heat up most if all other variables are constant. Therefore, when materials are being heated dielectrically in series, the one in which it is desired to generate the most heat should have the lowest dielectric constant and the lowest electrical capacity. This is just the opposite of what happens when a single material is heated dielectrically; then, the highest dielectric constant is desired.

Although the dielectric constant is independent of the thickness or size of a material, the capacity depends on the thickness and the cross-sectional area of the material. Therefore, these two dimensions must be taken into consideration when determining the capacities and voltage drops across the various parts of the trim sandwich.

The dielectric constant of a material may change with the frequency of the applied voltage; the temperature, moisture content, and impurity content of the material; and the pressure exerted on the material. The dielectric constant usually decreases with increasing frequency and increases as the moisture content of the material increases.

Power Factor

The percentage of electrical energy in the electric field that is changed into heat energy in a dielectric material upon reversing the field indicates the power factor. The power factor is a measure of the characteristic of a material to convert the electrical energy in the alternating electric field into heat energy.

The power factor is determined primarily by the linkages that exist between the molecules of a material. These linkages hold the molecules close together and cause friction when the molecules move. Therefore, when the molecules rotate in a reversing electric field, heat develops. A high power factor, then, contributes to greater heat in dielectric embossing.

The power factor of an insulating material increases as the moisture content of the material is increased, and may increase, decrease, or remain the same when the frequency of the alternating, electric field-producing voltage increases. The power factor also changes as the material heats during embossing so that complicated relationships occur during the embossing cycle.

Loss and Heating Factors

The power factor and dielectric constant of any insulating material may be multiplied together and expressed as a single factor called the loss factor. The loss factor of a material indicates the amount of heat that will be generated within it for any given voltage and frequency. The loss factor for materials which are successfully heated dielectrically is between 0.1 and 1.0 for frequencies above one megacycle.

The loss factor multiplied by the frequency is often referred to as the heating factor. The heating factor is an indication of how much heat is obtained at a given frequency.

It might be asked why higher frequencies are desirable if the dielectric constant decreases with increased frequency and the power factor also can decrease as the frequency is increased. The explanation is that the rate of change of dielectric constant or power factor with increasing frequency is not as great as the rate of change of frequency (Table I).

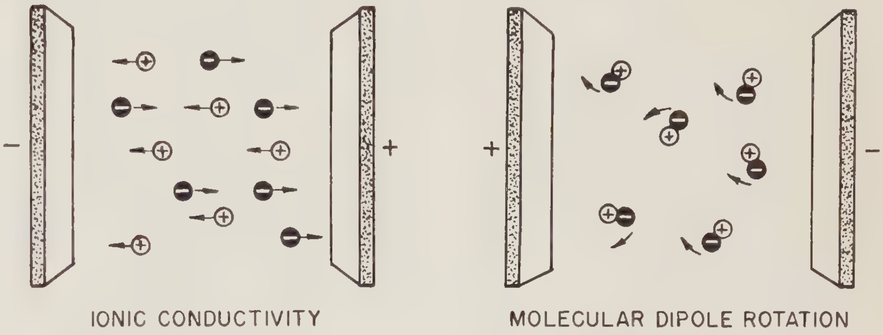


Fig. 4—The two basic dielectric heating mechanisms are ionic conductivity (left) and molecular dipole rotation (right). Molecular dipole rotation is illustrated in Fig. 2. Ionic conductivity is the reaction of ions within the trim material to the charged plates. The positive and negative ions are attracted to the oppositely charged plates. Since the charges on the plates are rapidly reversing in dielectric heating, there is much back and forth movement of the ions causing heat by friction.

Dielectric Strength

Another important factor controlling the use of insulating materials for dielectric embossing is the *dielectric strength*. The dielectric strength is simply a measure of how strong an electric field a material can withstand without rupturing electrically. The dielectric strength is equal to the number of volts needed to produce the strongest electric field that a particular thickness of a given material can withstand, and is expressed in volts per mil.

It is important to have a sufficiently high dielectric strength in all the materials and not to exceed the voltage limits of these strengths. If arcing or burning occurs, holes may be burned in the surface trim and there is danger of melting sections of the expensive metal embossing blades.

In selecting materials, such as adhesives, where a high loss factor is desired, the dielectric strength is often the limiting factor, since it almost always decreases as the loss factor increases.

The amount of heat developed in an insulating material when it is placed in an alternating electric field is given by the formula:

$$H = \frac{5}{9}(E^2FL)$$

where

H = heat developed (watts per cubic centimeter)

E = voltage gradient (thousands of volts per centimeter)

F = frequency of the alternating voltage (millions of cycles per second)

L = loss factor of the insulating material.

Problems of Applying Dielectric Heat to Trim Embossing

Before the dielectric heating process could be applied successfully to the embossing of automobile body interior trim assemblies, some important problems had to be solved and improvements made. Patent applications have been filed on various of the improvements mentioned in this paper. Not the least of the problems was the lack of knowledge in the industry of the electrical properties of trim materials in the megacycle frequency range and at temperatures up to 325°F.

Although it was realized that the dielectric constants and power factors of the insulating materials used in trim did vary with frequency and temperature and, equally important, with humidity, no quantitative data existed.

Even though it was known what electrical characteristics a trim material should possess to make it ideally suited to dielectric embossing, the facilities or experience to design these properties into the trim materials were not available. Besides, many other physical properties of trim materials had to be considered from a standpoint of cost, processing, and product service. Some of these other properties were in direct conflict with the necessary electrical properties for good dielectric heating.

A great deal of work has been carried out by the Fisher Body Division Process Development Department on the effects of humidity on the electrical properties of current trim materials. This study has shown the desirability of controlled humidity conditions in those plants using the dielectric embossing process. Excessively low humidity reduces the power factor and dielectric constant of most trim materials, which are hygroscopic, and requires higher embossing voltages. These higher voltages, in turn, result in more frequent arcing with consequently high scrap losses. Excessively high humidity, on the other hand, often results in too much heat being developed and in blistering of the vinyl coated trim materials.

An interesting sidelight of this investigation showed the need for more careful washing of fabrics to remove residual salts and ionizable substances introduced during bleaching and dyeing operations. Too great a concentration of ionizable substances caused localized overheating during dielectric embossing.

CHANGE IN DIELECTRIC CONSTANT WITH INCREASED FREQUENCY

FREQUENCY (megacycles)	DIELECTRIC CONSTANT	POWER FACTOR	LOSS FACTOR	HEATING FACTOR
1	2.9	0.17	0.49	0.5
5	2.6	0.09	0.23	1.1
10	2.4	0.07	0.17	1.7
50	2.3	0.05	0.11	5.5

Table I—The relationship of the characteristics of a typical trim material for different frequencies is shown in this table.

Dielectric Heating Used in Four Kinds of Fabrication

Four main classes of application of the dielectric heating process to trim fabrication have evolved in Fisher Body. They are:

- Normal embossing of thermoplastic trim materials
- Embossing of electrically conductive trim materials
- Embossing of trim materials with low loss factors
- Embossing compound curvatures.

Thermoplastic trim materials present no special problems to dielectric embossing. Thermoplastic materials now in use for interior body trim include unsupported vinyl film, vinyl coated fabrics, Vinyon and rayon fiber padding material impregnated with a thermoplastic resin, and paper boards laminated with polyethylene and coated with vinyl.

The normal construction for a trim panel consists of a laminated paper foundation board, padding, and a covering of vinyl coated fabric. Frequently a layer of unsupported vinyl film is dielectrically bonded onto the surface of the vinyl coated fabric to provide a contrasting color inlay.

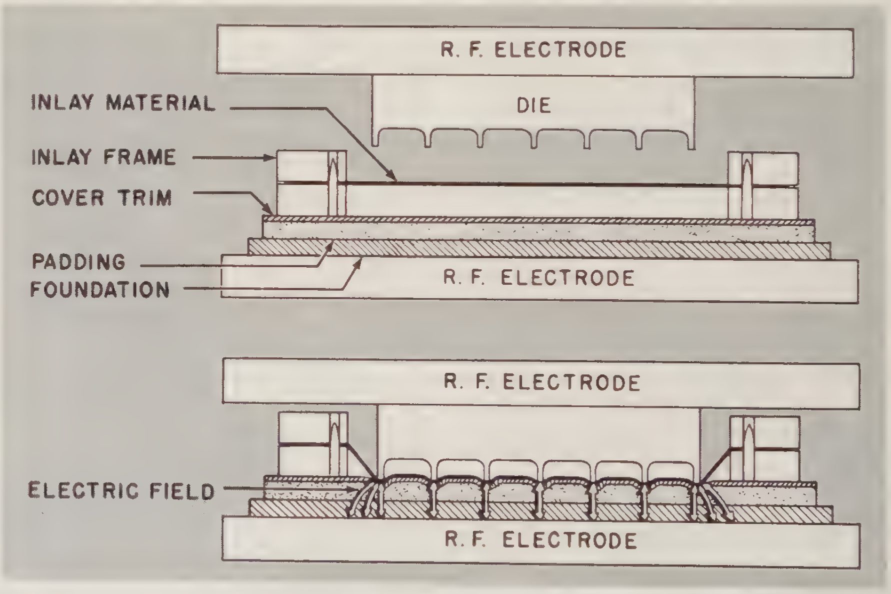


Fig. 5—Inlay trim embossing can accomplish two operations simultaneously. It can cut the inlay material to its desired shape and bond it to the trim sandwich. When the embossing press is open (top), the inlay material is suspended above the trim sandwich under tension in a frame. When the press is closed (bottom), the inlay simultaneously is cut and bonded to the trim. This is done by designing the embossing die to concentrate the electric field around the periphery of the inlay.

During the dielectric heating cycle, the thermoplastic resin in the padding melts and flows through the padding under the pressure of the embossing blades to bond the underside of the vinyl coated fabric to the top of the foundation board.

By a proper design of embossing blade, the electric field can be concentrated around the periphery of an embossed inlay design so that the unsupported vinyl inlay film which is held under tension by a retaining frame can be simultaneously cut to shape and bonded to the vinyl coated trim underneath (Fig. 5).

One of the newer types of trim materials available to the industry consists of a metallized Mylar film bonded onto a vinyl backing. By vacuum metallizing, a microscopically thin layer of aluminum is deposited on a Mylar film which is itself only 0.001 in. thick. This Mylar film is bonded to the thicker supporting vinyl with the metallized coating between them. The trim material gives the appearance of a highly polished sheet of metal, and is very useful for door kickpads which were formerly made from thin metal.

The metallized coating, even though it is extremely thin, acts as a conductor of electrical current and makes normal dielectric embossing techniques difficult.

If conventional dies were used to emboss the metallized trim, there would be excessive conduction of electrons in the coating and "tracking" and burning would occur (Fig. 6).

A second or "image" die is used to control the electrical field vertically and minimize the voltage gradient that would otherwise exist on the surface of the conducting trim material. The upper die, which is on the side of the metallized Mylar trim, makes contact with the entire surface of the trim. The image die (Fig. 6) is on the opposite side of the trim sandwich and controls the electric field to give the pattern of heat required.

Some padding materials used in embossed trim assemblies have relatively low loss factors. Foam rubber and felt are two such materials, and, while they have other desirable characteristics from a product quality or cost standpoint, their dielectric embossing characteristics are poor. A controlled amount of moisture is introduced into foam rubber and felt padding materials just before embossing. The moisture increases the ion content of the material and rapid heating takes place as a result of ionic conductivity.

Dielectric embossing of compound curvatures is encountered in the embossing of convertible top seams. These seams are blind sewn and then embossed for a water tight joint. Both the upper and lower embossing dies must be curved to fit the desired joint contour and very carefully matched to each other for an even pressure distribution. The choice of metal for the upper die must take into consideration the thermal conductivity of the metal as well as its chemical inertness.

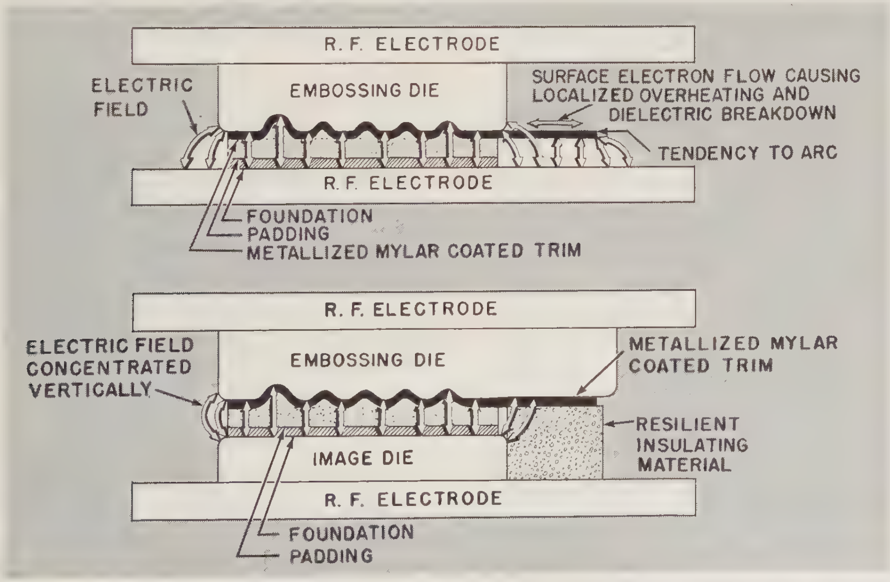

R. F. ELECTRODE
ELECTRIC FIELD
EMBOSSING DIE
SURFACE ELECTRON FLOW CAUSING LOCALIZED OVERHEATING AND DIELECTRIC BREAKDOWN
TENDENCY TO ARC
R. F. ELECTRODE
FOUNDATION
PADDING
METALLIZED MYLAR COATED TRIM
R. F. ELECTRODE
ELECTRIC FIELD CONCENTRATED VERTICALLY
EMBOSSING DIE
METALLIZED MYLAR COATED TRIM
RESILIENT INSULATING MATERIAL
IMAGE DIE
R. F. ELECTRODE
FOUNDATION
PADDING

Fig. 6—Dielectric embossing of conducting materials, such as metallized Mylar-film coated trim, requires the use of an image die to control the electric field. With a conventional embossing die (top), a voltage gradient exists on the surface of the conducting material, which causes localized heating and a tendency to arc. By using an image die (bottom), the electric field is controlled, and proper embossing conditions are assured.

This second point is important for the upper die comes in direct contact with the vinyl coated convertible top material. It was found that some metals stain the light colored top materials.

The proper selection and close control of dielectric embossing voltages, times, and pressures is essential to the quality control of embossed convertible top seams. The proper heat is that which will give a good vinyl-to-vinyl bond and result in a very small but uniform extrusion of vinyl from the joint on the visible side. Too much heat will result in excessive and ragged vinyl extrusion while too little heat will not give a good bond.

Cleanliness has been found to be of extreme importance in successful convertible top seam embossing. Even minute stray particles of metal act to concentrate the electric field and start arcs. Many of the newer trim fabrics are woven with periodic metallic threads and, in the cutting room, it is necessary to separate these materials from areas devoted to cutting convertible top materials.

Summary

The art of dielectric embossing of automobile body trim is not a static one. Changes are constantly being made in equipment, tooling, and processing techniques to accommodate new trim materials and new styling trends. That the dielectric heating process is flexible enough to keep pace with these changes is a tribute to its versatility and usefulness.

The key to the successful application of dielectric heat to body interior trim fabrication is a knowledge of the basic facts of dielectric heating and the behavior of a large variety of trim materials under the influence of high intensity, alternating electric fields.

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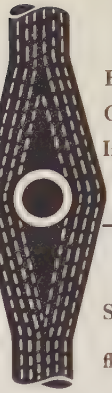
The Application of a Magnus Effect Flow Meter as a Laboratory Teaching Device



By **WESLEY J. TRATHEN**
General Motors
Institute

Seldom discussed fluid flow principle now visually demonstrated

In the majority of undergraduate fluid mechanics courses little emphasis is ever given to the Magnus effect, which occurs when a stream of fluid moves perpendicular to the geometric axis of a rotating body of revolution, such as a cylinder. This was the case at General Motors Institute. It was realized, however, that an introduction to the Magnus effect and its application would provide a firm foundation for understanding the fundamental principles of lift. As a result, a Magnus effect flow meter was developed for laboratory use. The flow meter allows a student to study, analyze, and use the principles of the Magnus effect. The flow meter also can be used to supply data for use in either calibration work or flow measurement.

In 1853 Professor Heinrich Magnus of the University of Berlin discovered an unusual phenomenon relating to a stream of air flowing around a single rotating cylinder. Professor Magnus discovered that when an air current was directed against a cylinder rotating in the same sense as the direction of air flow, the stratum of air nearest the surface of the cylinder revolved with the cylinder. The next nearest stratum also revolved, but at a slower speed. In a like manner, succeeding strata of air at progressively greater distances from the surface of the cylinder rotated at decreasing rates of speed until, at a certain distance, rotation of the air stopped.

Professor Magnus discovered further that if the flow of air impinging upon the cylinder was at a speed lower than the circumferential speed of the rotating cylinder, the streamlined flow was diverted in such a way that on one side of the cylinder the air was rarefied and on the other side the air was compressed (Fig. 1). These changes of pressure created a lateral thrust of power in a direction away from the high pressure side of the cylinder. Actually, the real direction of power was not always at right angles to the direction of the air current, but diverged in a measure dependent upon the speed ratio of air current to that of the circumferential speed of the rotating cylinder.

The phenomenon of a lateral thrust of power being created due to the action of an air current acting on a rotating cylinder has come to be known as the *Magnus effect*, after its discoverer.

Magnus Effect Applied to Ship Propulsion

The most spectacular application of the Magnus effect occurred in the mid 1920's when Anton Flettner, a German, applied the idea to ship propulsion (Fig. 2). Flettner fitted a 960-ton displacement schooner, the *Backau*, which had auxiliary power of approximately 200 hp, with two 42-ft masts. Bearings were mounted on each mast at the top and bottom to allow for free rotation of a 9-ft diameter, 3½-ton cylinder fitted over each mast. Each cylinder, which was rotated at a speed of 125 rpm by d-c motors, had a circumferential speed of approximately 60 fps. The circumferential speed of the cylinders was varied by suitable controls and was maintained at between 3½ to 4 times the linear speed of the wind.

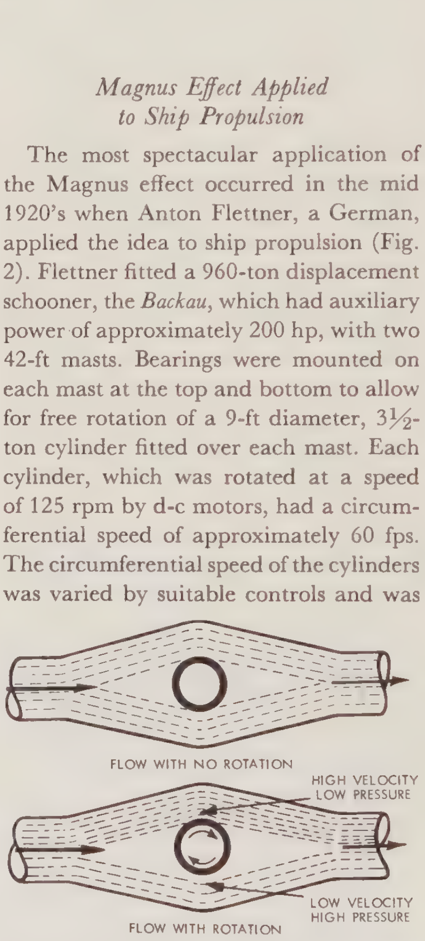


Fig. 1—Fluid flow about a non-rotating cylinder has a symmetrical streamline picture and pressure distribution (top). When fluid flows about a rotating cylinder, however, the situation changes (bottom). Because of viscosity, the fluid tends to move with the rotating surface and with a velocity diminishing with distance from the surface of the cylinder. On one side of the cylinder the velocity is directioned with the main stream velocity and on the opposite side the velocities oppose. This results in different net velocities, and consequently, different net pressures. This pressure difference and resulting lateral thrust are known as the *Magnus effect.*

The ship was propelled by means of altering the speed of either of the cylinders to change instantly the pressure exerted by the wind upon the cylinders. Each cylinder was provided with a disc of larger diameter at the top and bottom to increase the wind effect. Since the propulsion effort was dependent upon the ratio between the speed of the wind and the circumferential speed of the rotating cylinder, and since the circumferential speed could be kept constant by suitable controls and not allowed to rise above a fixed maximum, the wind pressure exerted upon the cylinders could rise only to a certain value—even in a hurricane. The cylinders continued to function in very high winds. In fact, when the wind grew stronger its effect on the cylinders was reduced automatically so that no great increase in pressure was noticed.

After sea trials were completed successfully, the *Backau*, which was then referred to as a rotor ship, was rechristened the *Baden Baden* and in 1926 made a round trip from Germany to New York. The following year a larger rotor ship, the *Barbara*, crossed the Atlantic ocean twice. While the rotor ships served to create a great amount of interest, little more was ever heard of them.

Magnus Effect Illustrated By Means of a Flow Meter

The Magnus effect and its application, while relating to an understanding of fluid flow about cylindrical objects, is seldom discussed and often ignored completely in undergraduate fluid mechanics courses. Such was the case at General Motors Institute, where only brief mention was given to the subject. It was

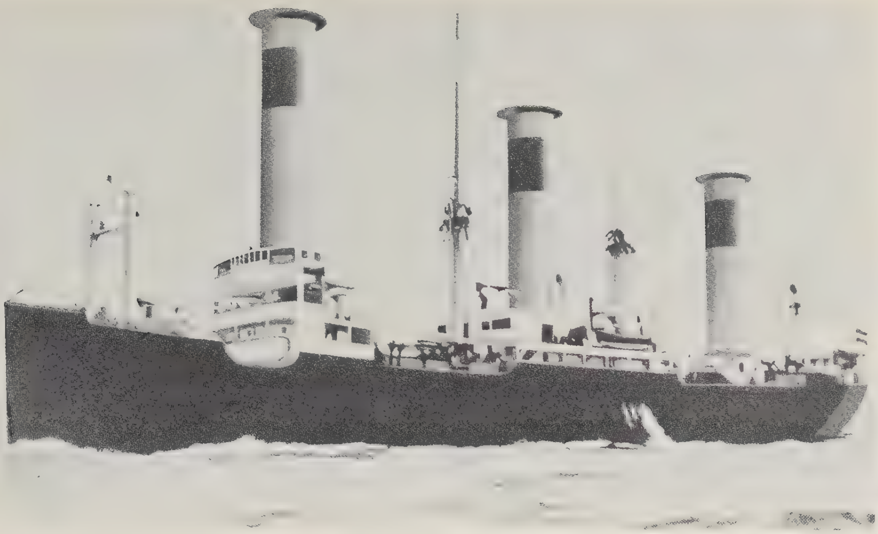


Fig. 2—The Magnus effect was applied during the 1920's to the development of rotor ships. Shown here is an artist's representation of the *Barbara*, the largest rotor ship built. This ship was propelled by three 23-ft diameter, 98-ft high cylinders which rotated at a speed of 150 rpm. A disc was fitted on the top and bottom of each cylinder to increase the wind effect. The *Barbara* crossed the Atlantic ocean twice during 1927. Since that time little has been heard of the rotor ship.

realized, however, that a broader knowledge of the Magnus effect would provide a good background for the study of fundamental principles of lift.

It was understood that possibilities of choice in practical laboratory applications of the Magnus effect would be limited. In view of this, it was decided to illustrate the Magnus effect by means of a flow meter (Fig. 3). A flow meter was selected because of its compactness as a piece of laboratory equipment, adaptability to illustration of fundamentals to students, and its ability to supply data for either calibration work or flow measurement.

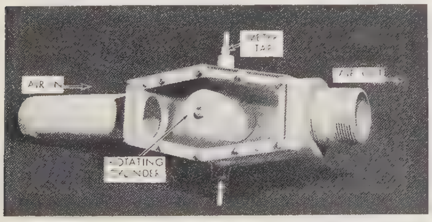


Fig. 3—The Magnus effect flow meter developed by General Motors Institute as a laboratory teaching device is designed to be mounted on a 2½-in. pipe with standard pipe threads. The cylinder is rotated by a 1/50-hp, 5,000-rpm motor. The main body and integral motor support are cast in one piece from aluminum. The flow cavity is machined all over to give a smooth passage and equal areas on both sides of the cylinder. Air is used as the fluid, giving a clean and convenient installation. The body casting has a front cover made of transparent plastic to allow students to see the cylinder rotation and predict the pressure differential from the velocities.

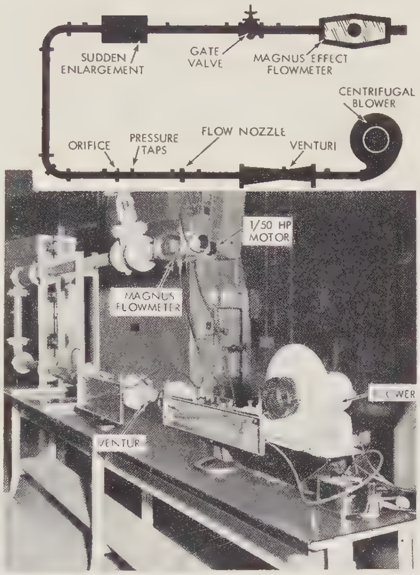


Fig. 4—The Magnus effect flow meter, shown here installed in an air flow stand, is used to illustrate Magnus effect fundamentals to students, and supply data for either calibration work or flow measurement. In addition, the performance of the Magnus effect flow meter can be compared to a venturi, since both meters are installed in the same line (inset).

The project was undertaken not to develop a flow meter of superior performance or of cheaper cost, but to serve as a means for allowing a student to study, analyze, and use the principles of the Magnus effect. As a laboratory device the Magnus effect flow meter has the following advantages:

- Minimum space requirements
- Low material cost
- Simple construction
- Sturdiness.

In brief, the laboratory procedure used at G.M.I. to illustrate the Magnus effect is to ask the student to examine the meter, noting flow direction and the direction of rotation of the drum. The student is then requested to predict the direction of the pressure differential across the meter after which he must confirm his predictions by operating the meter. The Magnus meter is then calibrated against a venturi placed in a connecting pipe (Fig. 4) by observing a series of simultaneous pressure drops across each meter.

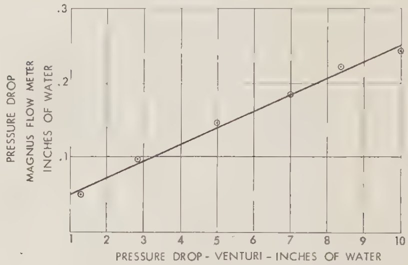


Fig. 5—One phase of the laboratory procedure used to illustrate the Magnus effect flow meter at G.M.I. is devoted to comparing its performance to that of a venturi. Shown here is a plot of simultaneous readings taken for both meters. The result shows linearity.

By equating the weight rates of flow and by using the known calibration of the venturi, the calibration constant for the Magnus meter can be obtained readily.

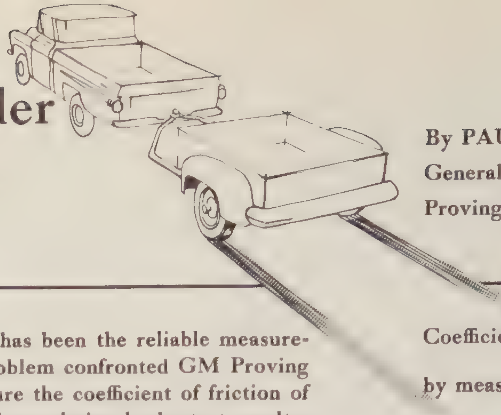
In addition to using the flow meter to illustrate the Magnus effect, the student also is able to compare the performance of the Magnus effect flow meter with a venturi. Simultaneous readings of the two meters are taken and plotted against each other, resulting in a linear function (Fig. 5). The observed pressure differential, with an average approach velocity of 25 fps, is 0.24 in. of water. These relatively small readings do not contribute to accuracy or sensitivity.

Conclusion

The Magnus effect flow meter is ideally suited as a laboratory teaching device. Reasonable in cost (the two main items being a fractional horsepower motor and a body casting) the flow meter has made it quite easy for students to understand the principles of the Magnus effect.

Engineering educators interested in obtaining further information on the use of the Magnus effect flow meter as a laboratory teaching device may write to General Motors Institute, Science Department, Chevrolet at Third, Flint, Michigan.

Simple Two-Wheel Trailer Measures Road Surface Coefficient of Friction



By PAUL C. SKEELS
General Motors
Proving Ground

Coefficient determined by measuring torque of sliding wheels

A problem frequently encountered by highway agencies has been the reliable measurement of road surface coefficient of friction. The same problem confronted GM Proving Ground engineers recently when it was desired to measure the coefficient of friction of various Proving Ground road surfaces for the purpose of correlating brake test results. To solve the problem Proving Ground engineers built a two-wheeled trailer which provides the means for measuring accurately the coefficient of friction value of any type of road surface. Designed for easy duplication by highway agencies and developed around specific design objectives, the trailer measures the friction force developed by braked wheels sliding on the road surface. This force is then converted into a friction coefficient value. The trailer is presently being used by various highway agencies to detect changes in road surfaces caused by the polishing action of tire wear.

HIGHWAY construction and maintenance agencies have been hampered in past efforts to obtain reliable measurements for the coefficient of friction, or skidding tendency, of road surfaces owing to a lack of suitable measuring equipment. Various methods of measuring the friction coefficient have been used. One of the most popular involved wetting a section of road with water, driving a car onto the wet section, locking the brakes, and measuring the distance required to stop from the point of brake application. The coefficient of friction was then calculated from the following relationship:

$$\mu = \frac{V^2}{30S}$$

where

μ = average coefficient of friction
V = initial velocity at which brakes were applied (mph)
S = stopping distance (ft).

Other methods for measuring the friction coefficient involved the use of decelerometers. The majority of methods used, however, have resulted in hazardous brake applications and an interruption of normal traffic flow. In addition, most of the methods lacked suitability for application on hills or curves.

A similar situation regarding measurement of road coefficient of friction existed at the GM Proving Ground, Milford, Michigan. About four years ago, however, GM Proving Ground engineers wanted to evaluate the coefficient of friction of Proving Ground road surfaces for the purpose of correlating brake test results. It was decided to use a somewhat new approach to road coefficient of friction measurement.

A passenger car was equipped with instrumentation to measure the torque reaction of the rear wheel brakes. The front wheel brakes were disconnected and the car towed behind a sprinkler truck (Fig. 1). The rear wheel brake reaction torque was read on a servo-type direct reading indicator calibrated to read pounds of force at the road surface. The car was towed over the wetted area with the rear wheels sliding and the force developed by the sliding wheels was observed. The coefficient of friction of the surface was then easily calculated, since the weight on the wheels, the wheelbase of the car, and the height of the tow attachment were known. The latter two factors were necessary in the calculation to correct for the weight transfer from the rear to front wheels when the brakes were applied.

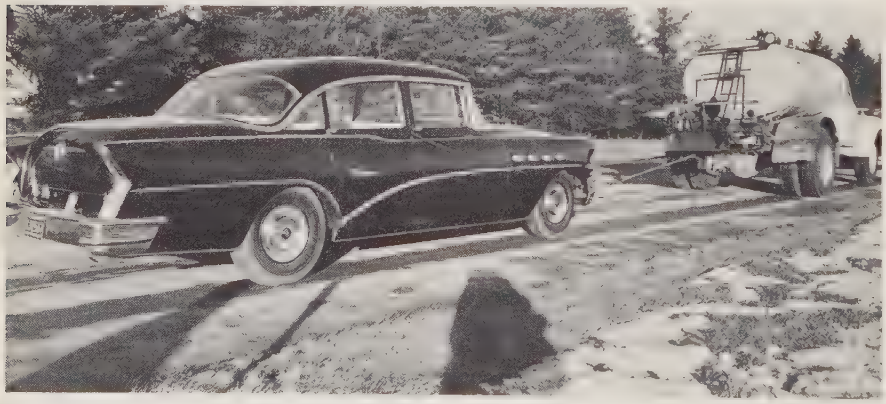


Fig. 1—An early method for determining road surface coefficient of friction at the GM Proving Ground was to tow a passenger car behind a water sprinkler truck, then brake the rear wheels of the car and measure the torque developed by the sliding wheels. Knowing this torque plus the weight on the rear wheels, the wheelbase of the car, and the height of the tow attachment provided the required information for determining the coefficient value. This method proved highly successful and created much interest among highway agencies.

This equipment proved entirely satisfactory for its intended purpose, and, after its existence became known[1], several highway agencies requested its use to determine the coefficient of friction of selected portions of highways. From contacts with highway agencies, it became apparent to Proving Ground engineers that there was a great amount of interest in road coefficient of friction, especially in the relationship of wet roads to accidents.

Special Trailer Developed to Measure Road Friction

Because of the interest indicated by highway agencies, Proving Ground engineers decided that a definite contribution to highway safety could be made through the design and development of equipment specifically intended for measuring the road coefficient of friction. Information obtained for the coefficient of friction between tire and road on various Proving Ground road surfaces would be of value to both automotive and highway engineers. In addition, it was further thought that if the intended equipment were

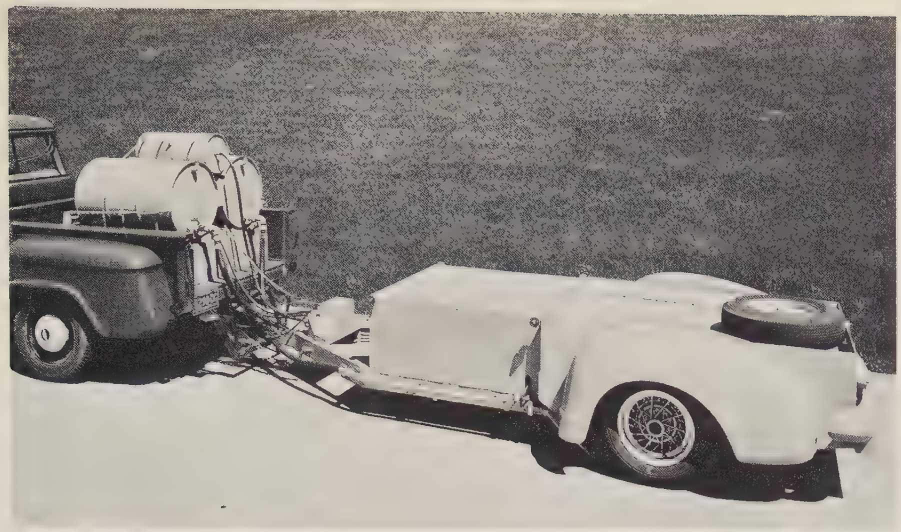


Fig. 2—The road surface coefficient of friction trailer developed by GM Proving Ground engineers is built around a used automobile chassis. Added to the re-worked chassis are a tongue, body, braking controls, road wetting equipment, and required instrumentation. The trailer, designed for easy duplication, provides an accurate and rapid measurement of the friction force developed when the trailer wheels are braked and allowed to slide over the road surface. The measurement obtained is then converted to a coefficient of friction value. To minimize brake overheating and improve brake drum cooling, wire wheels are used on the trailer. Mounted on the bed of the 3/4-ton pickup truck used to pull the trailer are two 50-gal water tanks (converted oil drums) which supply water for road wetting purposes. Mounted between the water tanks is a vacuum reservoir tank used to supply vacuum for operating a standard truck-type, combination hydraulic and vacuum power braking system used for the trailer wheels. The pickup truck also carries a water pump to re-fill the water tanks from streams or ponds when necessary.

judiciously designed it then could be duplicated easily by any highway agency interested in using it on their own road system.

With these ideas in mind, Proving Ground engineers inaugurated a design and development program aimed at producing road coefficient of friction measuring equipment which would satisfy the following ground rules:

- Overall design of equipment and related components to be as simple as possible and based on the use of relatively inexpensive and readily available parts
- Data provided to be accurate and available in either indicated or recorded form
- Operation of equipment to require a minimum of manpower
- Equipment to have the capability of being operated in a completely safe manner and at speeds which would provide a minimum of interference with the normal flow of traffic in order to make it useful on city streets and heavily traveled roads.

Based on past experience, it was known that the most accurate method for determining road surface coefficient of friction was to measure the torque resulting from braked wheels sliding on the road surface. It was decided that the most effective way to utilize this principle would be through the use of a two-wheeled trailer pulled by a light-weight truck, each being equipped with suitable controls and instruments. Using this as a starting point, Proving Ground engineers designed and developed a road coefficient of friction trailer (Fig. 2) which fulfilled the design objectives.

Trailer Construction

The first design objective—construct the trailer from relatively inexpensive and readily available parts—was achieved by deliberately designing the trailer to be constructed of salvage material. This makes it possible for any highway agency to obtain materials easily in order to build duplicate equipment.

The trailer was constructed from a 1950 Buick automobile chassis. A Buick chassis was selected since it uses a torque tube drive with coil springs—an arrangement well suited for the installation of strain gages to measure brake reaction. Any Buick chassis built since the end of World War II is suitable for the trailer.

The front section of the chassis was removed and to the remaining section were added a body, tongue, braking and road wetting controls, and instrumentation required for the measurement of brake reaction (Fig. 3).

The body of the trailer serves as the means for providing the necessary static weight on the trailer wheels. The body, made of steel boiler plate, brings the rear wheel weight up to 2,000 lb, which approximates the weight on the rear wheels of present day passenger cars. Material other than steel could be used, so long as the necessary weight is provided.

Braking and Water Controls

In keeping with the first design objective, components for the braking and road wetting controls were kept as simple as possible. The complete rear running gear of the chassis was retained along with the original brakes. The only modifications were in the method used to apply the trailer brakes. This was accomplished by using a standard truck-type power brake unit (Hydrovac), a component easily obtained from any automotive supply house. (The Hydrovac is a combined hydraulic and vacuum power braking system which uses the vacuum created in the engine intake manifold as an operating force.)

Braking of the trailer wheels is controlled manually from the cab of a 3/4-ton pickup truck used to pull the trailer. A single lever operated from the cab of the truck (Fig. 4) controls a timer which opens water valves to direct water flow immediately ahead of each tire and applies the brakes a short time later.

The water supply for road wetting purposes is obtained from two 50-gal tanks (converted oil drums) mounted on the bed of the truck. Each drum feeds water by gravity flow to one wheel of the trailer through a large diameter plastic hose. Water flow through each line is controlled by a low restriction ball valve. The valves are actuated by vacuum driven windshield wiper motors which are controlled from the lever used to apply the trailer wheel brakes.

The water supply control system also was designed for simplicity and easy duplication. Small vacuum diaphragms, obtained from a 1957 Oldsmobile heating system, provide the means for operating two windshield wiper motors. These motors, which obtain their vacuum for operation from the line feeding the Hydrovac unit, turn the handles of the two ball valves.

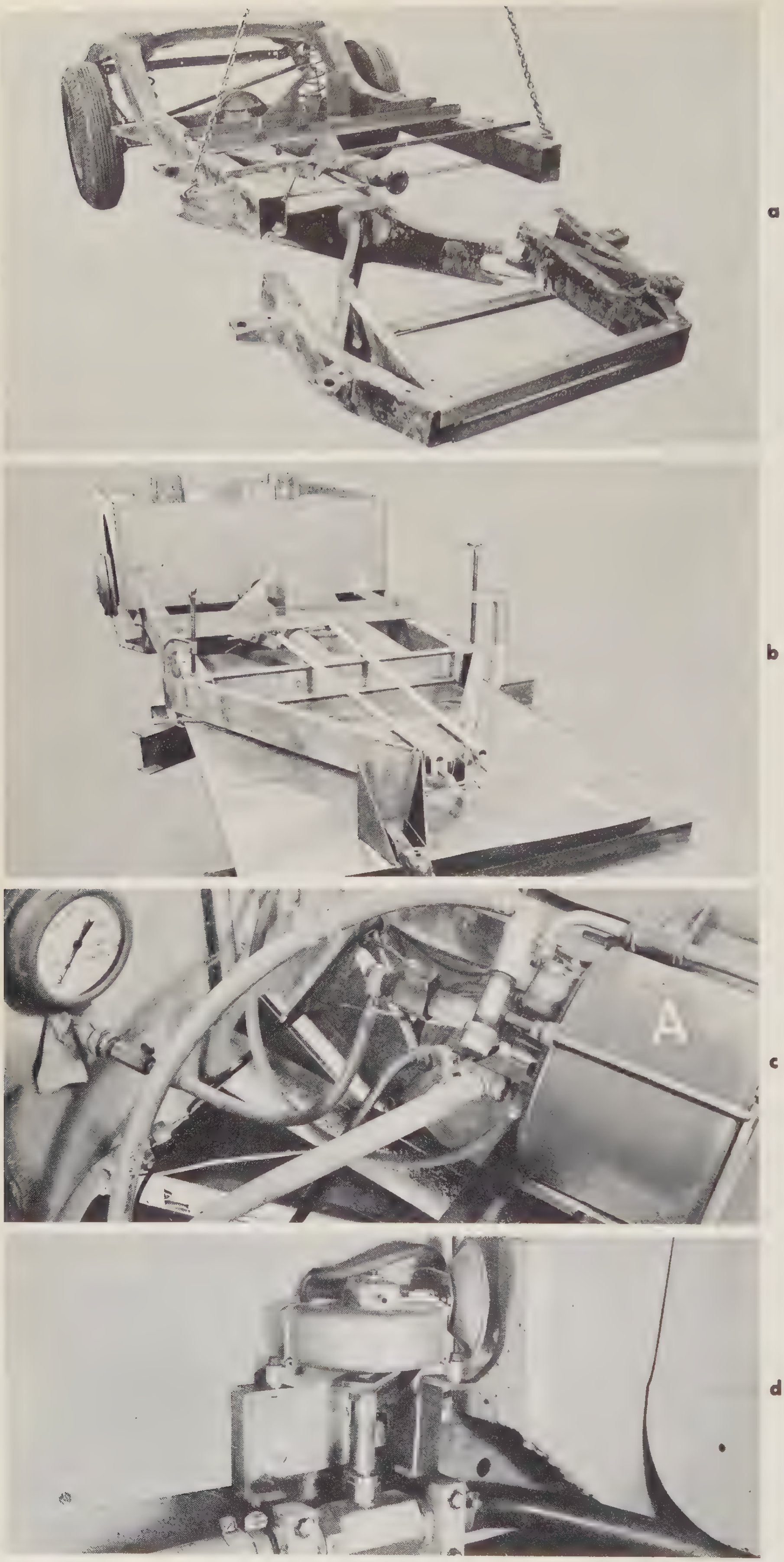


Fig. 3—The basic component of the road surface coefficient of friction trailer is a 1950 Buick automobile chassis. The drive shaft contained in the torque tube has been removed, the front end of the frame cut off (a), and the side rails strengthened by the addition of channel members. Added to the re-worked chassis are a tongue and body (b). The final stage in the trailer assembly consists of mounting various items of control equipment in the trailer body (c), one of which is a standard truck-type power braking system *A*. Windshield wiper motors (d) operate ball valves to allow water to be sprayed directly in front of the trailer wheels. Required plumbing for the vacuum and water system also is mounted in the body of the trailer.

A developmental problem occurred with this phase of the water control system. Gate valves were originally used. These valves, however, would stick occasionally because of the wedge shape of the moving part of the valve. This problem was partially solved by using small adjustable stops on the valve handle to prevent the valve from closing tightly. This resulted in a small trickle of water through the closed valve, but assured consistent operation. The gate valves, however, were replaced with ball valves as a water conservation measure.

Instruments Measure Sliding Wheel Torque

To achieve the second design objective—accuracy of data—required the development of a reliable method for measuring the torque resulting from the braked wheels sliding on the road surface and then converting this measurement into readable form by suitable instrumentation.

To measure the torque developed by the sliding wheels, strain gages were attached to the top and bottom of the torque tube at a point immediately ahead of the differential. This point of attachment was used for the following reasons. When the brakes are locked to slide the trailer wheels, the axle housing tries to rotate because of the torque developed. This rotational tendency is restrained by the torque tube, which extends from the nose of the differential housing to a frame cross member located near the front of the trailer. The forces developed result in a bending moment in the torque tube which is greatest at the point of attachment to the differential.

Measuring the tire-to-road force by measurement of brake reaction torque is much superior to the more obvious method of vehicle drawbar force. This is because the drawbar method introduces errors due to the inertia of the trailer and gravity components when operating on hills. These errors can easily amount to 25 per cent and are not present if brake torque is used on the measurement.

The strain gages were connected as a full Wheatstone bridge and were completely protected against mechanical and water damage. Wires were connected from the gages to a terminal box located at the front of the trailer.

The instrumentation first used to read the force developed by the sliding wheels was a direct reading, servo-type, self-balancing potentiometer mounted on the dashboard of the truck. This indicator, constructed in the Proving Ground laboratory, did not produce a permanent record and the reading was recorded manually by an observer. While not suitable for indicating the transient force involved in the incipient slide condition, it proved highly successful for indicating the force developed during the sliding condition. The reading was taken during the first one or two seconds following the start of wheel slide. During this period of time the reading was relatively stable if the road surface was uniform. This indicating instrument has recently been replaced by a modified commercial recording instrument so that a permanent record may be obtained (Fig. 4).

Since the recorder operated from a direct current input, it was necessary to supply the strain gage bridge from a stable voltage source. In addition, a secondary calibrating system was supplied so that the overall sensitivity could be checked at will. With the a-c servo system used on the direct reading indicator this was not necessary, since voltage and frequency fluctuations in the power supply were compensated for inherently.

The value indicated by the recording, servo-type, self-balancing potentiometer is a torque which is related to a friction force F_f developed by the sliding trailer wheels. The torque is produced by F_f acting at the road surface. In practice, the meter is calibrated directly in force units.

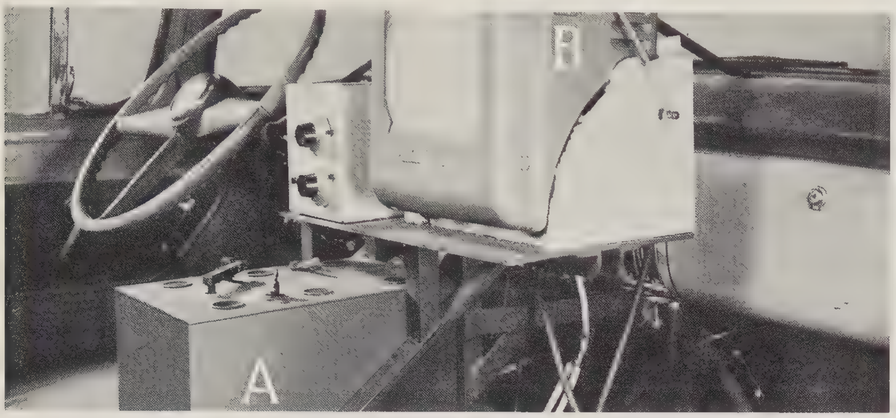
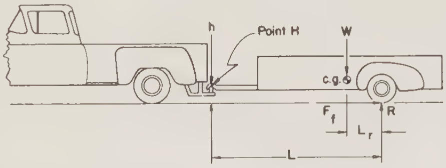


Fig. 4—Trailer braking and road wetting are controlled from the same control box *A* mounted on the seat of the pickup truck between the driver and observer. A single lever operated by the observer opens the water valves and applies the trailer brakes a few seconds later. The friction force created by the braked wheels sliding on the road surface is read and recorded on a servo-type, self-balancing potentiometer *B*. The reading, which is taken during the first one or two seconds following the start of wheel slide, is converted to coefficient of friction by means of a previously prepared table listing coefficient values for each increment of friction force. A standard trailer braking control (not visible), mounted on the steering column, is provided for emergency use. This control does not allow the driver to modulate the trailer brakes. By judicious handling of the control, however, the driver can obtain considerable braking assistance under emergency conditions.

The road coefficient of friction for any friction force is calculated as follows:



Let

μ = road coefficient of friction
F_f = friction force developed by sliding wheels (lb)
R = total trailer wheel vertical reaction force (lb)
W = weight of trailer (lb)
W_s = static weight on trailer wheels (lb)
h = height of trailer hitch above ground level (in.)
L = distance from trailer hitch to trailer axle (in.)
L_r = distance from trailer center of gravity to trailer axle (in.).

The road coefficient of friction is equal to

$$\mu = \frac{F_f}{R}. \qquad (1)$$

A summation of horizontal moments M_h about point H gives

$$\sum M_h = W(L - L_r) - RL - h(F_f) = 0. \qquad (2)$$

Solving equation (2) for the reaction force R gives

$$R = \frac{W\,(L - L_r)}{L} - \frac{h}{L}\,(F_f). \qquad (3)$$

Substituting this value for reaction force R into equation (1) gives the following expression:

$$\mu = \frac{F_f}{W\left(1 - \frac{L_r}{L}\right) - \frac{h}{L}\,(F_f)}. \qquad (4)$$

The term

$$W\left(1 - \frac{L_r}{L}\right)$$

is the static weight W_s on the trailer wheels. Substituting this information into equation (4) gives the following equation for road coefficient of friction:

$$\mu = \frac{F_f}{W_S - \frac{h}{L}\,(F_f)}. \qquad (5)$$

For the Proving Ground trailer, the height h of the trailer hitch above ground level is 10 in., the distance L from the trailer hitch to the trailer axle is 120 in., and the static weight W_s on the trailer wheels is 2,000 lb. For the Proving Ground trailer, therefore, the road coefficient of friction is equal to the friction force F_f divided by the quantity 2,000 minus 1/12 of the friction force. For Proving Ground test work, a table is prepared which shows the coefficient of friction for each increment of friction force indicated by the recorder.

Instrument Calibration

To assure accuracy and reliability of data a suitable method had to be developed for calibrating the instrumentation used to indicate friction force. The calibration method developed uses an electrical drawbar (Fig. 5) inserted as a connecting link between the tow truck and trailer.

Calibration is accomplished by first operating the tow truck and trailer on a road surface known to have a reasonably high coefficient of friction. The trailer wheels are then braked to skid and the trailer pulled by the truck. The drawbar and the indicator, mounted in the cab of the truck, are read simultaneously taking readings ranging from a low drawbar force up to the maximum obtainable before sliding the trailer wheels. Up to

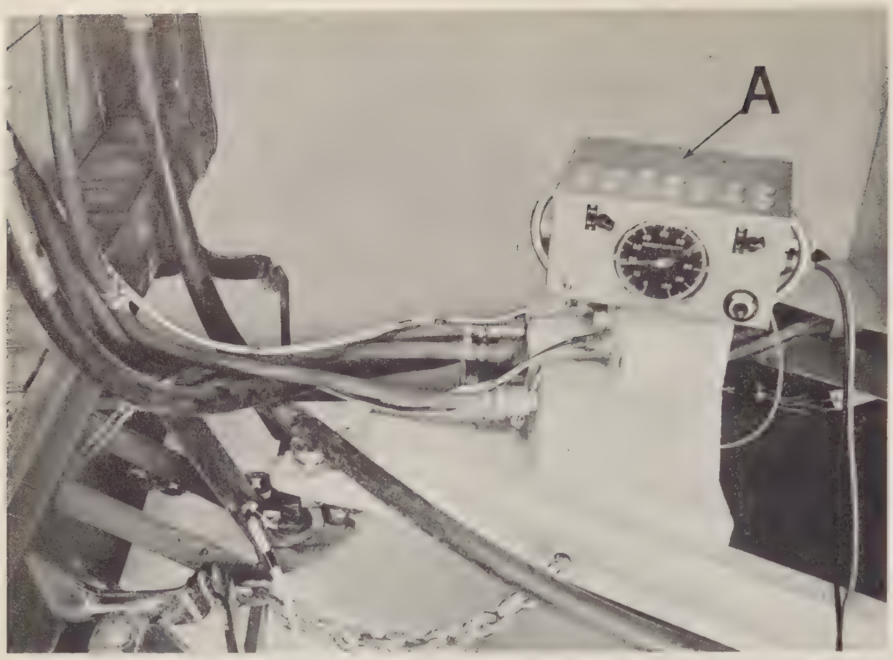


Fig. 5—To calibrate the friction force indicating equipment, an electrical drawbar *A* is inserted as a connecting link between the truck and trailer. The drawbar, which provides the means for measuring the horizontal force applied to the trailer, is made in the form of a trailer hitch ball. The drawbar contains strain gages mounted on the shank and arranged to measure sheer in the column under the ball so that the point of application of force at the ball end is immaterial to the reading obtained.

the point of wheel slide, the drawbar force applied to the trailer is equal to the friction force F_f developed by the tires. By applying equation (5), a calibration curve is drawn which is used to convert the friction force read on the indicator directly to coefficient of friction.

Since the standing height of the rear axle affects the torque developed in the trailer axle housing for a given friction force F_f, it also affects the calibration of the indicator. Generally, it has not been found safe to assume that tires of different makes have the same standing height, even though they may be the same size. To assure accuracy, the friction force indicator is calibrated each time the trailer tires are changed, unless the tires are from the same batch.

Field Tests Establish Operating Procedure

Prior to conducting formal road coefficient of friction measurements, a series of field tests were made to determine the best operating procedure. These tests served to introduce a few improvements in the equipment. The basic design of the trailer was apparently sound, however, since no serious problems developed and continuous all-day operation was found perfectly feasible. As many as 500 individual measurements were made during one day's operation.

The field tests also served to indicate that the remaining design objectives had been achieved. It was found that measurements could be easily obtained with two operators—a driver and an observer. Also, the equipment was found capable of operating in a completely safe manner at speeds which would not interfere with normal traffic flow. The trailer tracked satisfactorily up to speeds of 70 mph and a constant speed could be maintained with the trailer wheels sliding at speeds between 50 mph and 60 mph.

Because the force measuring system was designed to be virtually unaffected by gravity components, the trailer performed well on hills and curves. A change in velocity of the trailer during the time a measurement was made did not affect the accuracy of the reading, since trailer inertia does not contribute to the measured braking torque. This change in velocity was rarely over one or two mph, even for a 50-mph measurement run.

Field Tests Also Studied Incipient Coefficient of Friction

One phase of field testing was devoted to a study of incipient coefficient of friction—that coefficient existing just prior to wheel slide. These tests were conducted with a recording oscillograph connected to record friction force, since the servo-type indicator had too slow a response for the measurement of incipient coefficient.

In general, the incipient coefficient was higher than the coefficient during wheel slide. Also, the rapid change of wheel speed occurring as the wheels changed from a rolling to a sliding condition tended to give a high peak reading at the moment the wheels began to slide because of the polar moment of inertia of the wheels, axles, and differential. This peak reading was not apparent on the direct reading indicator but was readily recorded by the oscillograph.

Difficulty was experienced in pinpointing the incipient coefficient of friction on the first recordings obtained (Fig. 6—top), and it was realized that the method used for reading the incipient values was not completely accurate. Improved instrumentation was then installed to enable more accurate measurements to be made (Fig. 6—bottom).

To analyze properly the records obtained, it became necessary to use a comprehensive mathematical analysis to evaluate the effect which wheel inertia had on the recordings. Because the mathematical treatment was rather unwieldy and difficult to apply to each individual reading, a graphical method was devised which duplicated the mathematical treatment to a satisfactory degree of accuracy. A comparison between the mathematical correction and the graphical correction for many points over a wide speed range (Fig. 7) showed that the two methods gave results which check within about 10 per cent. Since this correction was applied to a reading in the range of 2,000 lb, the average difference was between one and two per cent, which was considered satisfactory.

To obtain further experience with the road coefficient of friction trailer, a series of tests were run to evaluate variations between different makes of tires at various speeds and on various surfaces (Fig. 8).

Wet Pavement Offers Best Conditions for Friction Measurements

Once the field tests were completed and the operating procedure established, formal road coefficient of friction tests were then conducted.

Since the majority of serious skidding problems occur on wet pavements, most of the tests have been conducted under wet road-surface conditions. Tire wear during these tests has not been a problem, unless the trailer brakes cause the

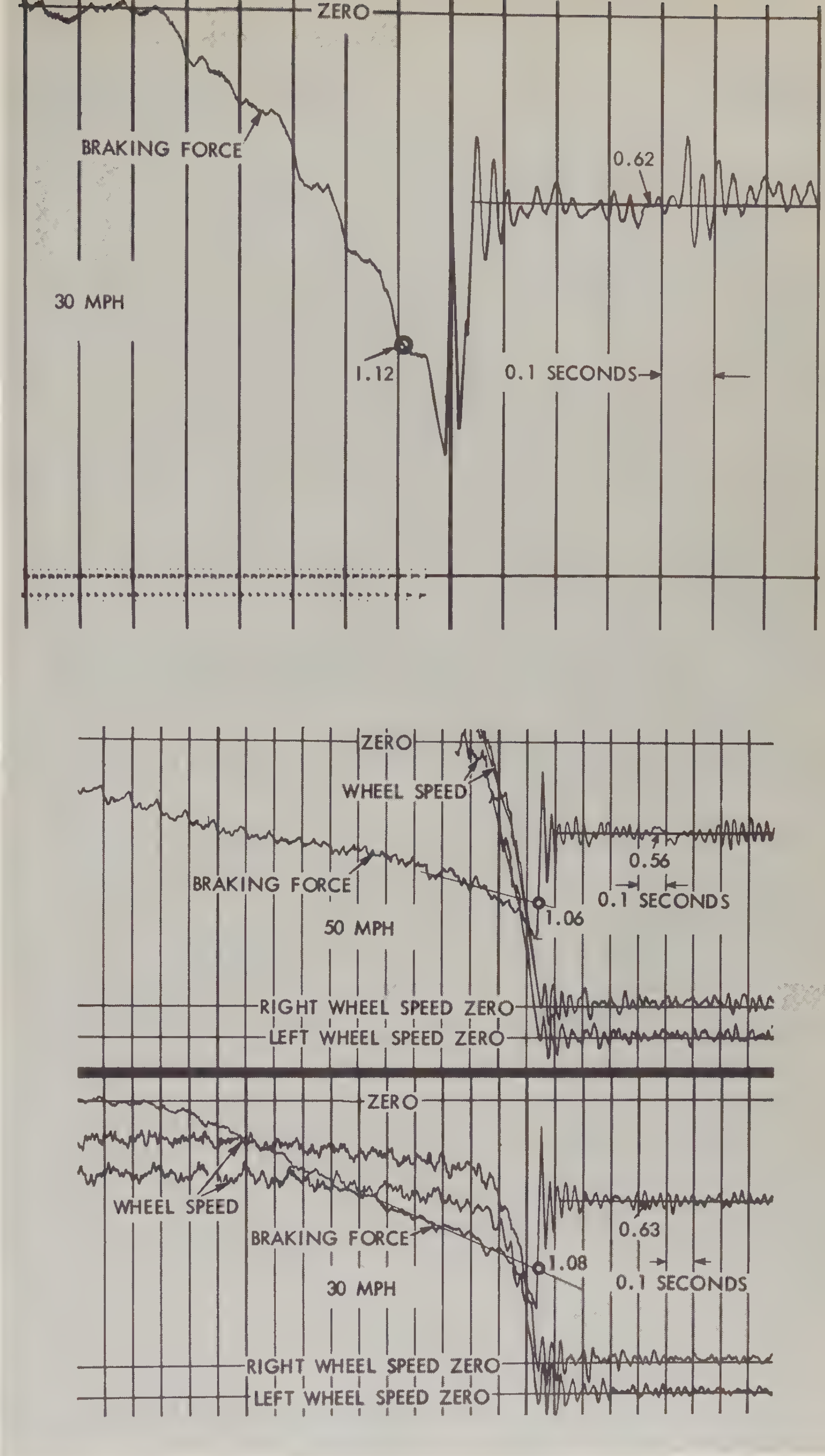


Fig. 6—To study incipient coefficient of friction, a recording oscillograph was used in place of the direct reading, servo-type indicator. One of the records obtained from the recording oscillograph is shown here (top). The horizontal line at the top represents zero braking force on the trailer wheels. As braking force develops, the line moves downward. The pips at the bottom of the record each represent 1/10 of a wheel revolution on one wheel. As these pips broaden, decreasing wheel speed is indicated. As they disappear, wheel slide is indicated. The braking force can be seen to increase at a nearly linear rate until just before wheel slide, at which time the slope increases. As the wheels start to slide an abrupt drop in braking force is noted and a 35-cycle per second oscillation develops.

The sliding coefficient is derived from the average force immediately after the wheels begin to slide and is represented by the straight horizontal line drawn through the oscillating trace indicated by the arrow and the number 0.62. The incipient coefficient is harder to pinpoint since the brake force trace is the result of two distinct input force systems: the force developed at the tire-road contact point; and the force due to the kinetic energy withdrawn from the wheels as their velocity is reduced to zero. At the time this record was taken, it was common practice to read arbitrarily the low point immediately preceding the spike which occurs just prior to wheel slide. This point also was indicated by the pips at the bottom of the record. These pips indicated nearly normal wheel velocity at this time. This point is shown by the number 1.12.

It was realized that this method of reading incipient coefficient of friction was not accurate. As a result, improved instrumentation was installed and additional records obtained (bottom). The record shown here was obtained at two speeds—30 mph and 50 mph, all other factors being equal. In taking these readings, tachometer generators were installed on each trailer wheel, and their output recorded along with braking force. Since it was desired to observe the instant each wheel reached the sliding condition, the zeros were offset and are indicated by the two bottom horizontal lines. The wheel speed traces are somewhat irregular because of rotational oscillation of the tachometer generators. The top record shows that both wheels slid at nearly the same instant, and that the drop in brake force occurred when the wheel speed reached zero. To evaluate the effect of wheel inertias on these records, a comprehensive mathematical analysis was made, based on these wheel speed data. The mathematical treatment, however, was rather unwieldy and difficult to apply to each individual reading. A graphical method was then devised which duplicated the mathematical method to a satisfactory degree of accuracy. The graphical method assumed that the rate of rise of the brake force curves was essentially linear, and that this linear trend continued to the point of wheel slide. The incipient coefficient of friction was derived from the value which the extrapolated curve had at the moment the slide occurred. This value is illustrated by a small circle on both the top and bottom records. The relationship of calculated to graphically determined points justifies the use of the graphical method (Fig. 7).

wheels to slide before the water is applied, or hold the wheels in slide after the water is shut off. With the brakes in proper adjustment, a thousand or more tests can be made on one set of tires with good, consistent results.

Some tests have been conducted on dry pavements. The measurements obtained, however, have not been consistent because the indicated coefficient usually dropped rapidly during the period of wheel slide. This made it difficult to assign a single value to the reading. The cause of this change is not thoroughly understood, but it could be the result of temperature build-up at the contacting surfaces, or a lubrication of the area by soft or torn-up rubber. This condition does not prevail on wet pavement slides.

Dry sliding has resulted in very short tire life, since each slide marks the pave-

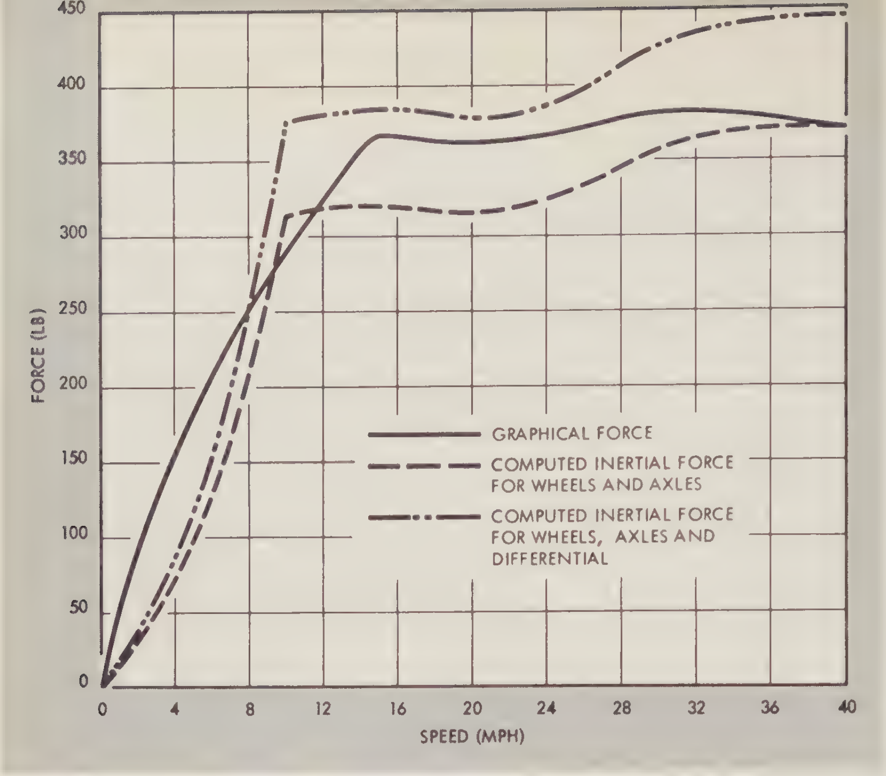


Fig. 7—The relationship of calculated to graphically determined points for one series of tests, covering a wide speed range, conducted to study incipient coefficient of friction shows the near agreement of the results.

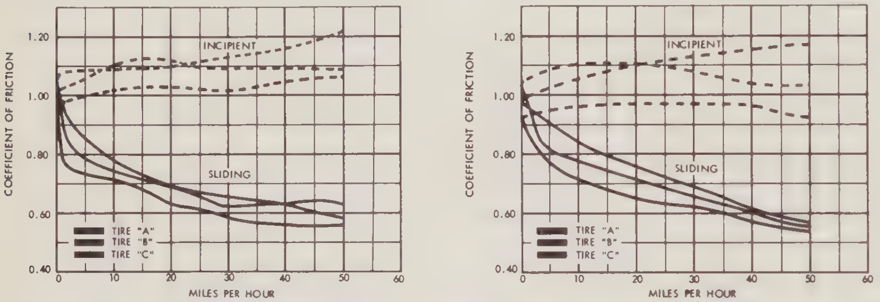


Fig. 8—To evaluate variations between different makes of tires, coefficient of friction tests were conducted on three brands of tires at various speeds on two different road surfaces. Both incipient and sliding coefficients were measured. The results summarized in the record at the left indicate a very rapid drop in coefficient between zero and five mph, with a less rapid drop as speed increases. It will be noted that in the 40 mph to 50 mph region the curves flatten out. To verify this trend, additional testing work at higher speeds must be performed. On a different road surface, this tendency was not demonstrated. For the curves shown in the record at the right, obtained on a different road surface, it will be noted that the incipient coefficient seems to have a rising tendency with speed. Although this tendency is not great, it has been evidenced on all tires run to date. It can be said that incipient coefficient does not fall off with speed as sliding coefficient does.

ment and produces a flat spot on the tire. Also, the surface of the tire becomes rough and its coefficient characteristics change rapidly. Although these conditions might not be severe on polished pavement, it can be stated safely that dry road surface coefficient measurements are not as accurate or as easy to make as those on wet surfaces. The work done on dry pavements has indicated that the coefficients measured during the early portion of each slide do not drop off drastically with speed as they do on wet pavements.

To date, road coefficient of friction measurement work performed by GM Proving Ground engineers has been done with commercially available tires made of GR-S rubber (government rubber-styrene). Although the tires have a significant effect on the coefficient values obtained, the operating procedure usually involves a survey of a certain road system with the object of pin-pointing slippery sections. On this kind of survey the absolute value is not of prime importance. However, if many highway agencies make coefficient of friction measurements, a standard test tire would be desirable to give results which can be compared on a common basis.

Summary

The GM Proving Ground road coefficient of friction trailer, developed around definite design objectives, has served to fulfill a long-felt need in highway safety planning. The trailer has been duplicated and is presently being used by several state highway departments.

Road surface coefficient of friction is affected not only by the type of road building materials used but also by the constant polishing action of the surface caused by automobile and truck tires. In studying this latter effect, the trailer has been specifically useful to highway engineers in detecting changes in road surfaces due to polishing action, thereby leading to resurfacing of slippery sections of heavily traveled roadways and, as a result, making them safer for the motoring public.

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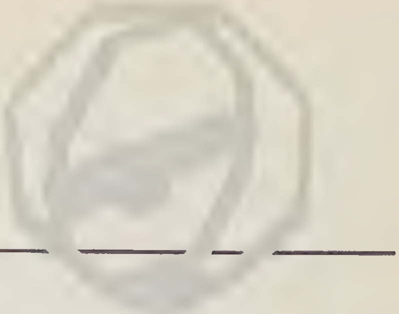
New Gyroscope Developed as an Aid to Teaching in the Field of Physics and Allied Sciences



Gyroscopic principles have been studied by men for many years, but it was not until the early part of the twentieth century that practical devices based on the principles of spinning rotors began to be developed.

Practical applications of the gyroscope have ranged from the early gyroscopic compass to the recent inertial guidance mechanisms for guided missiles. These recent developments in the guidance field have focused increased attention on the gyroscope and its theory.

The basic principle of the gyro is that of centripetal force. A rotating body or mass about an axle produces a force which resists extraneous forces, thereby permitting the axle of the rotating body to remain fixed in its own reference point or line in space. This force term is called *gyroscopic inertia* or *rigidity*.

All gyros have, in addition to the axle of the rotating body, or rotor, vertical and horizontal axes which must be mutually perpendicular. All gyros react 90° from the point of initially applied force. This action is called *precession*.

Gyros are classified into two categories: *captive*, or having one degree of freedom; and *free*, or having two degrees of freedom. The captive gyro eliminates the vertical axis and is free to precess about the horizontal axis when torque is applied in a horizontal or azimuth plane affecting angular displacement. Therefore, an output, or precession, occurs in the freedom axis (horizontal) from angular velocity about the input axis, or vertical axis, through the center of the gyro base. This kind of device is classified further according to the kind of restraint which is applied to the developed, or output, torque.

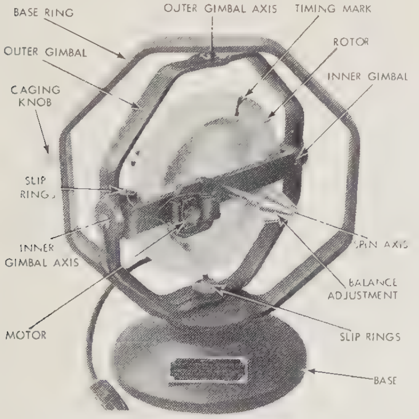


Fig. 1—MITAC, the classroom gyroscope, is constructed to permit experimentation with both captive and free gyros. The overall device is 15 in. high, 13 in. wide, and weighs 17 lb. The six-lb rotor is one in. thick and has a six-in. diameter. It is driven at 150 rpm by an electric motor which operates on 110-115 v, a-c current.

A *rate integrating* gyro is derived from the captive gyro by adding viscous restraint to provide an opposing torque proportional to angular velocity. This adds to the torque caused by accelerating the inertia of the inner gimbal and wheel.

Another configuration of the captive gyro occurs when the developed torque is opposed by inertia, viscous damping, and spring torque. In this form, the captive gyro displays an angular position output for an angular rate input, and often is called a rate gyro.

Captive gyros are used in stable platforms (inertial guidance systems) for simplicity and reliability. Rate integrating gyros are used in flight controls.

The free gyro represents a stable reference and, except for bearing friction and other extraneous forces, will maintain a fixed orientation with respect to inertial space regardless of the motion of the body to which it is fixed. Specific applications of the free gyro include temporary stable references, lead computation in gun sights, and torque measurements.

A gyroscope is difficult to understand until seen in action. With the growing commercial emphasis on the gyroscope, there has been an increased need for gyros designed for educational purposes. To meet this need AC Spark Plug Division, in cooperation with the Massachusetts Institute of Technology, designed and developed MITAC, a classroom gyroscope (Fig. 1).

This classroom gyroscope was designed as a versatile device for the purpose of demonstrating both captive and free gyros with various combinations of inertial, viscous, and spring restraint. MITAC can demonstrate visually such phenomena as precession, gyroscopic rigidity, and nutation. It also can reproduce the effects of a rate gyro, integrating gyro, and displacement gyro for proving mathematical computation.

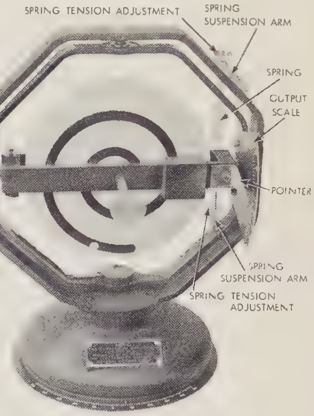
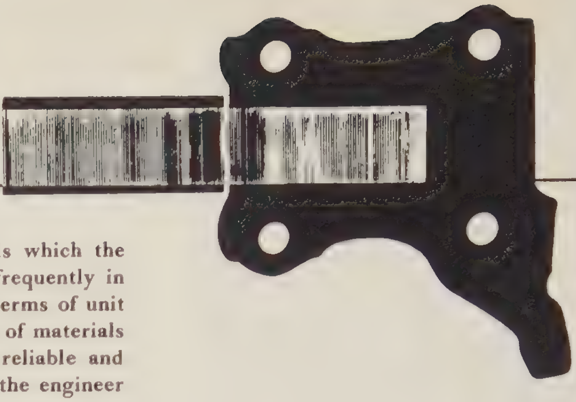


Fig. 2—By opposing the torque developed in a captive gyro by inertia, viscous damping, and spring torque, a rate gyro can be formed. This illustration shows a rate gyro application of the MITAC, which employs some of the accessories included in the classroom gyroscope kit.

The MITAC, which also can be used in research work, provides the student and the researcher with a unit of instrument quality which gives high accuracy while being unhampered by the delicacy, compactness, and expense associated with gyroscopes designed for specific, precision functions. Included with the gyroscope are restraints, pick-off scales, and other accessories necessary to conduct many experiments and demonstrations involving gyroscopic principles (Fig. 2).

The classroom gyroscope is being produced by AC Spark Plug Division, Flint, Michigan. It is available to educators and researchers at a nominal price. Included with MITAC is a handbook dealing with the theory of operation, experiments, and mathematical computations.

Some Applications of Stress Analysis Techniques in Improving Casting Designs



Load and deformation are two terms in the field of strength of materials which the student studies in the classroom and which the practicing engineer uses frequently in product design. Load and deformation can be measured and expressed in terms of unit stress and unit strain. A knowledge of the stress and strain characteristics of materials are important to the engineer in designing a product that will be both reliable and economical. Present-day techniques for measuring strain on a part enable the engineer to make a stress analysis of the part and thus recommend the optimum design. The results of such a stress analysis often are intriguing and they may suggest a different design or a different material than was previously proposed. At Central Foundry Division, stress analysis techniques are applied in a stress laboratory and in a specially equipped test vehicle to achieve better casting designs.

IMPROVING the design of iron castings and developing new applications for foundry products are objectives which underlie the work of engineers, technicians, and others at Central Foundry Division. This GM Division manufactures grey iron, malleable iron, and ArmaSteel* castings for use in passenger cars, trucks, Diesel engines, and non-vehicle applications. Stress analysis of casting designs is an important part of the product development to meet these objectives.

Stress analysis is the resolution of stresses occurring on a part due to its service function, and it is used for the determination and improvement of the mechanical strength of the part.

Product development includes both the improvement in casting quality and the search for new casting applications. It often means the accumulation of proof that the casting process can produce a better part at a lower cost than can many other fabricating methods.

Stress analysis at Central Foundry is conducted with the aid of the facilities of a stress laboratory and a specially equipped test vehicle. Experimental determination of the stresses usually involves the application of either the *brittle lacquer* or the *bonded wire strain gage* techniques to measure the deformation or unit strain of a part. Hooke's Law, expressing the fundamental relation between stress and strain, is then used to calculate the stresses. This law states that stress (unit load expressed in lb per sq in.) divided by strain (unit deformation expressed in micro in. per in.) equals the modulus of elasticity of a material. The three test materials most often encountered have the following values for modulus of elasticity:

grey iron	15,000,000 psi
malleable iron	25,000,000 psi
ArmaSteel	27,000,000 psi.

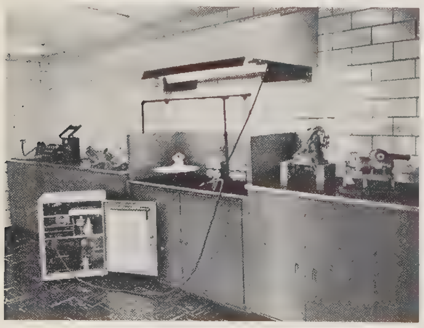


Fig. 1—The stress analysis laboratory at Central Foundry Division includes spray equipment for making brittle lacquer coatings on test parts, as shown in this view of the *build-up room.*

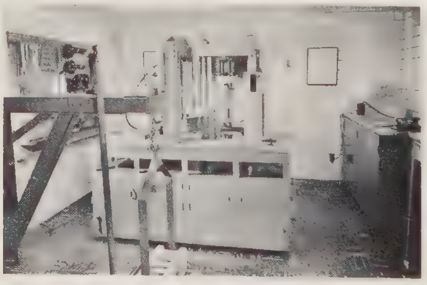


Fig. 2—The stress analysis laboratory also includes a *stress room* where testing is conducted under controlled temperature and humidity conditions. This room contains a 100-ton ram for load tests, a drying oven for curing coating and gale cements, and various other test fixtures.

Two Methods Used For Strain Measurement

The stress laboratory is made up of two rooms: a *build-up room* and a *stress room.*

The build-up room is used for preparing the parts to be tested (Fig. 1). It includes equipment such as:

- Down draft spray booth
- Special spray equipment for coating a test part with brittle lacquer
- Storage area for test equipment and materials.

The stress room is maintained at a controlled temperature and humidity because of the sensitivity of the materials used in stress analysis work. A specially constructed stress table is located in this room upon which all fixtures for tests are mounted. Hydraulic jacks, adapters, and pumps of various capacities are provided to simulate loading conditions on test parts (Fig. 2).

Brittle Lacquer Method

The brittle lacquer technique for measuring strain requires that the test part be coated with lacquer which becomes brittle when it hardens. When a strain of a selected magnitude is produced on the part, the lacquer cracks forming line patterns. These patterns indicate both the general location and direction of the strain, and from this information, a quantitative value can be obtained.

When preparing a part for test, it is necessary that the part be absolutely clean. After cleaning, a special aluminum lacquer undercoating is sprayed on the part. The undercoating aids in the determination of the thickness of the brittle lacquer coating which is subsequently applied to the part by repeated passings of a spray gun.

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*ArmaSteel is a trade name denoting a graphitic, steel-like material which consists of a matrix of sorbitic pearlite and spherodized cementite in which small nodules of carbon are imbedded.

By **LOUIS H. RAVITCH**
Central Foundry
Division

Using experimental measurements of strain in Hooke's Law

A calibration bar is prepared and coated in the same manner. The bar is used to determine the sensitivity of the brittle lacquer coating. After the part and the bar have been coated, they are placed in the drying oven for approximately 18 hours. After curing in the oven, the part is ready for test, and the coating on the bar is ready to be calibrated.

The calibration bar is mounted as a cantilever beam in a special loading fixture and subjected to a fixed deformation at its free end. As the load is applied gradually on the bar, a series of cracks appears in the brittle lacquer (Fig. 3). The point on the bar where the last crack appears is marked and this represents the sensitivity of the brittle lacquer coating.

The bar is removed from the loading fixture and placed in a calibration scale where the last crack mark is read in terms of micro in. of strain per in. This value is the indicated strain to initiate cracking.

To illustrate how the strain readings are used by the engineer in studying a casting design, an example of a U-bolt anchor plate for an axle application can be studied. The part was cast of Arma-Steel and was designed to accommodate two U-bolts and nuts to secure the rear spring to the axle. The casting was coated with brittle lacquer and the calibration bar indicated that the lacquer would crack at a strain of 400 micro in. per in. (Fig. 3). The coated part was placed in the test fixture in the laboratory and simulated service loads were applied (Fig. 4). As the torque loads on the part were increased, cracks in the lacquer appeared in several areas and in several patterns (Fig. 5).

One of the purposes of this test was to determine whether the U-bolt anchor plate would safely withstand a torque loading of 75 ft-lb on each of the four U-bolt nuts during assembly by the truck manufacturer. Therefore, loadings in increments of 25 ft-lb were applied during the test. A load of 25 ft-lb applied to each of the four nuts caused the first cracks in the lacquer to appear. The areas of these first cracks (Area 1) were the areas of maximum strain on the bracket, since they were the first to show crack patterns. By referring to the calibration bar for this test, these first cracks also meant that the value of the strain in these areas was 400 micro in. per in. Loadings were increased to 50 ft-lb and 75 ft-lb which produced additional patterns of cracks in the brittle lacquer, designated by Area 2 and Area 3, respectively. The stresses in the casting were determined by the following calculations in accordance with Hooke's Law:

$$s = \varepsilon E$$

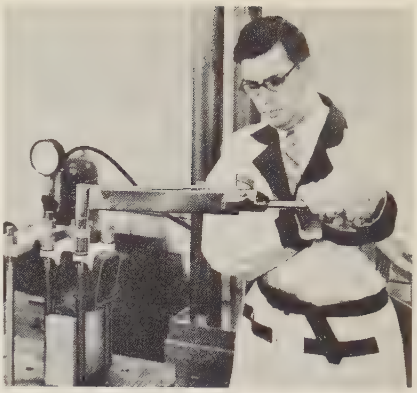


Fig. 4—A typical stress laboratory set-up for making a brittle lacquer test is illustrated above. A U-bolt anchor plate has been coated with lacquer and has been placed in a test fixture simulating the actual installation on a vehicle. Loads are applied with a torque wrench.

where

s = unit stress (psi)
ε = unit strain (in. per in.)
E = modulus of elasticity (psi).

Therefore,

$$s = (0.000400)\ (27{,}000{,}000)$$
$$s = 10{,}800 \text{ psi.}$$

Thus, the stress level in the areas where the cracks first appeared was 10,800 psi at the applied torque of 25 ft-lb. The stresses in the second and third areas resulting from applied torque loads of 50 and 75 ft-lb were calculated as:

$$\left(\frac{50}{25}\right)(10{,}800 \text{ psi}) = 21{,}600 \text{ psi}$$

$$\left(\frac{75}{25}\right)(10{,}800 \text{ psi}) = 32{,}400 \text{ psi.}$$

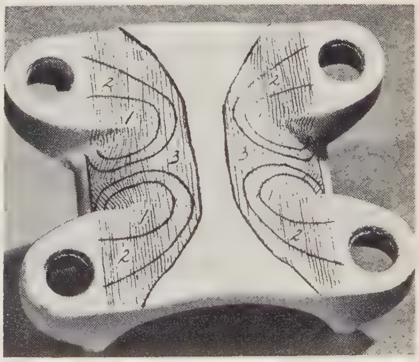


Fig. 5—The brittle lacquer test of the U-bolt anchor plate (Fig. 4) produced the results shown above. The cracks were marked with ink to be more visible in this illustration. The first crack patterns to appear were those marked *1*. These are the areas of maximum strain on this casting. A load of 25 ft-lb on each U-bolt nut produced the cracks in area No. *1*. As the loads were increased to 50 ft-lb and 75 ft-lb, cracks appeared in the areas marked *2* and *3*, respectively.

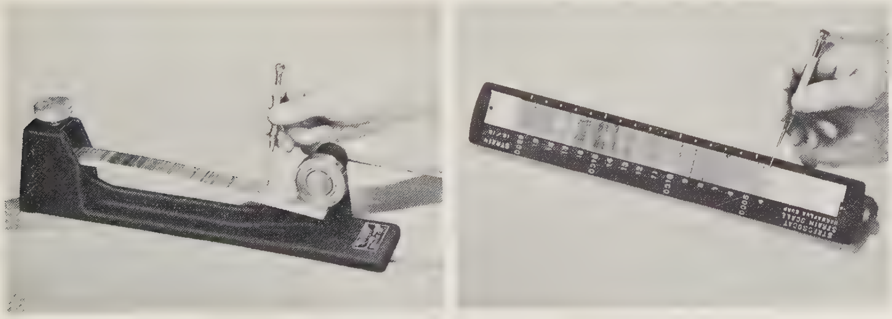


Fig. 3—A calibration bar is used in the brittle lacquer technique for measuring strain. The bar is coated with lacquer and is oven-dried at the same time as a test part to assure similarity of coatings and accurate calibration. The bar is placed in the load fixture, (left) and is gradually loaded to a fixed amount at its free end. As the bar is loaded, cracks in the lacquer appear. The bar then is placed in a calibration scale (right) and the value for the strain (micro in. per in.) is read at the point where the last crack appeared on the bar, as indicated. In the example above, the scale indicates that cracking of the lacquer will start at a strain of 400 micro in. per in. This means that a movement of four ten-thousandths of an in. per in. on the test part will crack the lacquer coating. With this information about the part under test, the stresses can be calculated.

To complete the test, torque loads beyond 75 ft-lb were applied. The U-bolt anchor plate did not yield indicating that it was more than adequate for the assembly requirement.

This information from the brittle lacquer tests aided in the redesign of the U-bolt anchor plate (Fig. 6). The original design, cast of malleable iron, weighed 2.25 lb while the new design, cast of GM 86M ArmaSteel, weighed only 1.30 lb. The ribs were moved from the outside of the casting to the inside to place them in compression rather than in tension. A center hole was eliminated. The resulting casting proved to be 35 per cent stronger than as previously designed.

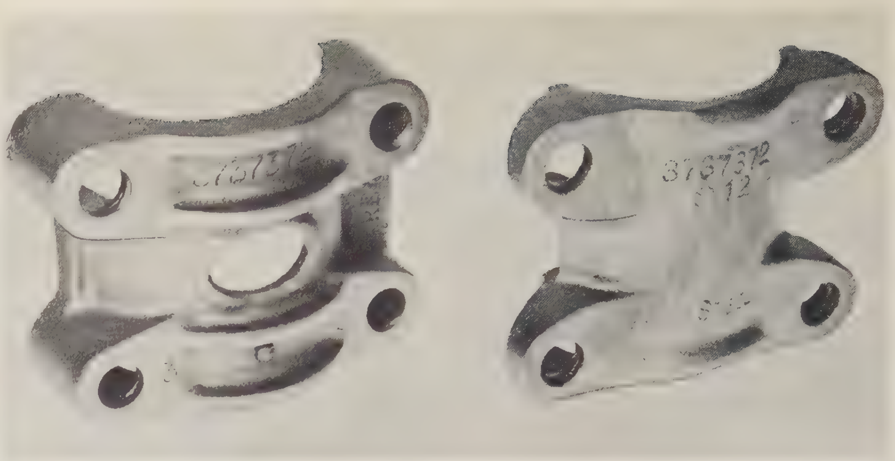


Fig. 6—The former design (left) of the U-bolt anchor plate for an axle application is compared with the redesigned version (right) which evolved through the use of stress analysis (Fig. 5) and the application of foundry and casting design principles. In the revised design the ribs were moved from the outside to the inside to place them in compression rather than in tension. The center hole was eliminated. The lugs gripping the axle were cut back angularly for gussetting effect. The heavy metal sections between the ears were removed resulting in a more uniform metal thickness. The redesigned casting was 35 per cent stronger and 41 per cent lighter than the former casting.

Strain Gage Method

The brittle lacquer technique for strain measurements is generally adequate for many routine applications in the Central Foundry laboratory. For more extensive stress analysis studies, however, the resistance wire, or strain gage, techniques also are employed. The brittle lacquer coating aids in locating the areas of maximum strain, and the strain gages then are oriented in these areas to obtain measurements.

Basically, a strain gage is a piece of one-mil wire in the form of a grid consisting of parallel loops cemented between two pieces of thin insulating material (Fig. 7). The strain gage is bonded with cement so firmly to the test part that it actually becomes an integral part of the test piece. When loads are applied to the test part, therefore, the strain gage is subjected to exactly the same loads both in magnitude and direction.

The strain gage is connected to a Wheatstone Bridge circuit which measures its electrical resistance so that any change in the diameter of the wire in the gage caused by compression or tension strains brings about a change in resistance in the circuit. This change in electrical resistance, caused by strain in the part, and consequently in the gage, is measured directly in micro in. per in. of strain by means of a special electrical calibration instrument called a *strain indicator* (Fig. 8). As in the case of the brittle lacquer technique, the stress on the part is calculated by means of Hooke's Law.

Dynamic Testing by
Specially Equipped Vehicle

To supplement the static testing facilities described above, Central Foundry also conducts dynamic testing of casting designs in a specially equipped vehicle containing portable electronic equipment. Engineers can mount a test part in position on the vehicle, a station wagon, to obtain the stresses actually encountered in normal road operation of the car.

The equipment in the test car consists of the following units: two pre-amplifiers, a two-channel recorder, an oscilloscope, a torque meter, and a power source and converter unit to operate this equipment. A switching panel also is installed with this equipment so that any of the gages located in different positions on the test part can be dialed into the circuit for recording purposes (Fig. 9).

The amplifier units are used to amplify, for recording purposes, the very minute electrical signal obtained from the gage. One amplifier is used for the gages on the test part, the other is used on the gages in the torque meter.

The torque meter, mounted on the drive shaft, is a slip ring, strain gage unit used to obtain the actual torque loads under service conditions. Mounted on the car instrument panel is a torque meter dial which indicates these torque loads directly in ft-lb.

The two-channel recorder is an ink pen unit used to obtain a permanent, strip-chart recording of these strain measurements for test data comparisons. One side of the strip-chart records the torque data while the other side records the strain data. Therefore, it is possible to compare strain on test parts under duplicate service conditions evaluating their design characteristics.

The oscilloscope is used to view the

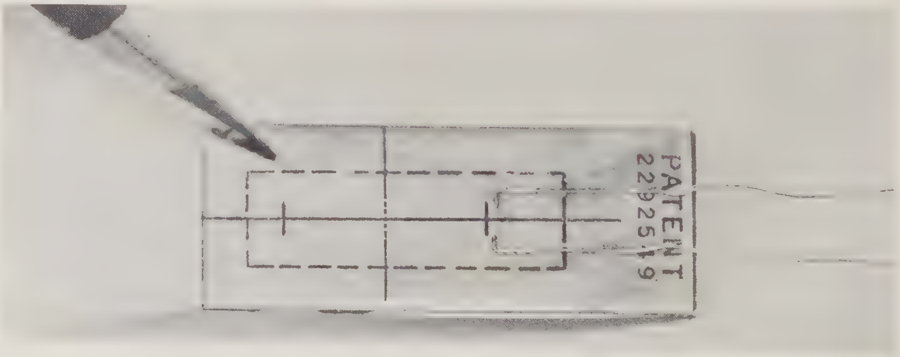


Fig. 7—The strain gage consists of a piece of one-mil wire arranged in a grid of parallel loops. The grid is cemented between two pieces of thin insulating material and the gage can be cemented firmly to a surface of the part being tested. A change in the length or diameter of the wire caused by strains on the part produces a change in electrical resistance in the wire. The resistance is measured and calibrated to read the unit strain directly in micro in. per in.

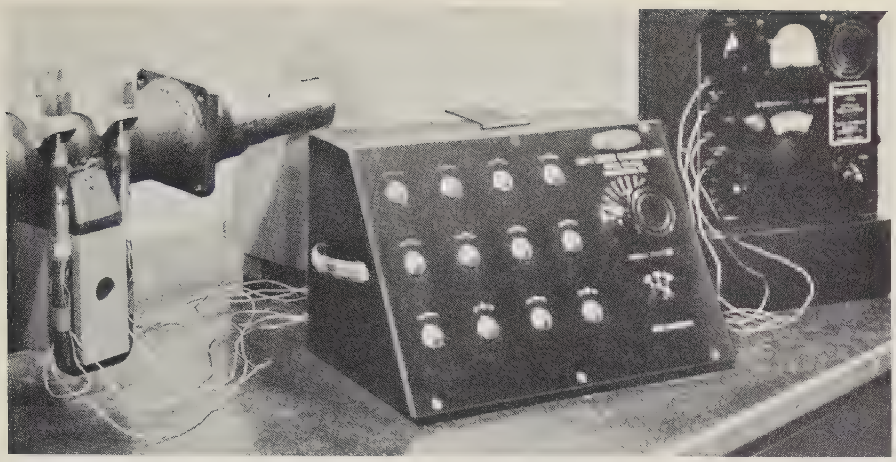


Fig. 8—Measuring unit strain with strain gages is illustrated in another application on the rear axle, U-bolt anchor plate assembly. Here the strain gages were cemented to the U-bolts to obtain service loads on the casting using assembly torque data. These calculated service loads were used to correlate load-deflection data on the test part for evaluation purposes. The strain gage wires were connected to a Wheatstone Bridge balancing unit, for switching purposes, and then connected to the strain indicator from which the magnitude of strain was read.

strain phenomena as they occur on the part at the same time that they are recorded on the strip chart. By means of calibrations on the oscilloscope screen, the magnitude of these strains can be obtained. These data may also be photographed for permanent records by means of a self-developing camera mounted on the front of the instrument.

Two storage batteries connected in parallel with the regular car battery are used to power this equipment. A converter is used to change the battery direct current to alternating current.

Before this dynamic testing facility was adopted, only those tests which could be reproduced in the laboratory with simulated service loads obtained by test fixtures were conducted. By means of brittle lacquer coatings on these test castings, the maximum stressed areas were located as indicated by dead loads applied on the part. These predetermined loads in many cases, were obtained from specifications supplied by the user of the casting. However, it was realized that these loads, as applied in the laboratory on simulated test setups, did not necessarily represent true operating conditions. Thus, the test car makes it possible to obtain additional data for casting design evaluation.

Stress Analysis Aids Product Design

The results of the stress analysis work, whether done in the laboratory or on the test car, have the primary purpose of serving as a guide to the product engineer. In the design and manufacture of cast products, there are a number of fundamental principles which designers follow. Some of these are reinforced by data from stress analysis, and the stress laboratory engineers often work in close contact with the product engineers in the design of a casting. These design principles can be summarized as follows (Fig. 10):

- Keep material away from the neutral axis for maximum strength
- Keep plates in tension and ribs in compression wherever possible
- Use smoothly tapered sections to eliminate high stress concentrations.

In addition, some foundry principles are (Fig. 11):

- At adjoining sections, avoid sharp corners and abrupt section changes by the use of fillets and blending radii
- Maintain casting sections as uniform in thickness as possible
- Design ribs and brackets for maximum effectiveness.

Examples of parts which were redesigned through the use of stress analysis

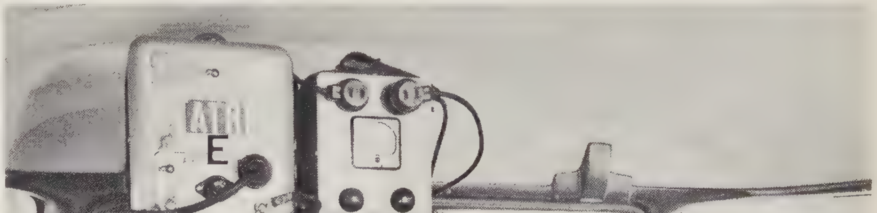
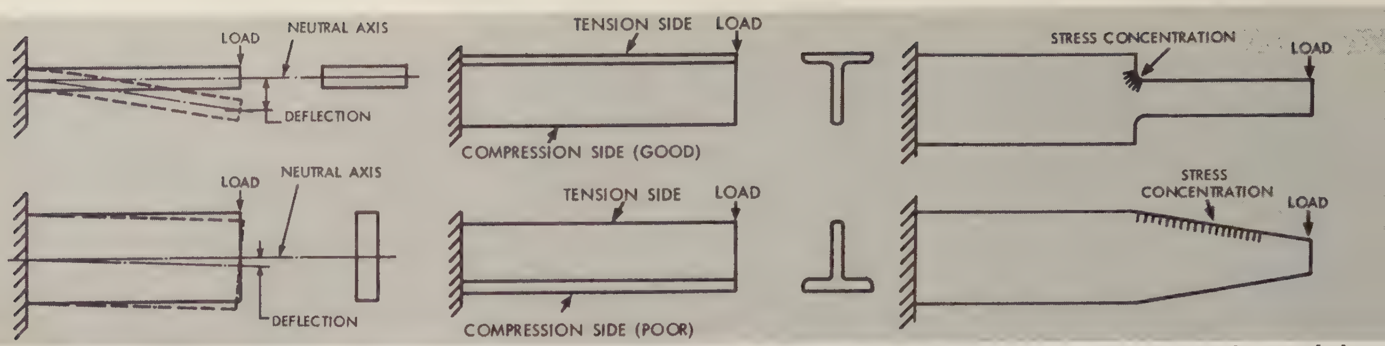


Fig. 9—Central Foundry Division's test car contains the following electronic equipment: two preamplifiers *A*, a two-channel recorder *B*, an oscilloscope *C*, and a switching panel *D* provided to connect the instrumentation with any one of several gages which might be placed in different test locations on the vehicle. The equipment is powered by two storage batteries connected in parallel with the regular car battery. An inverter *E* is included to provide the necessary alternating current.

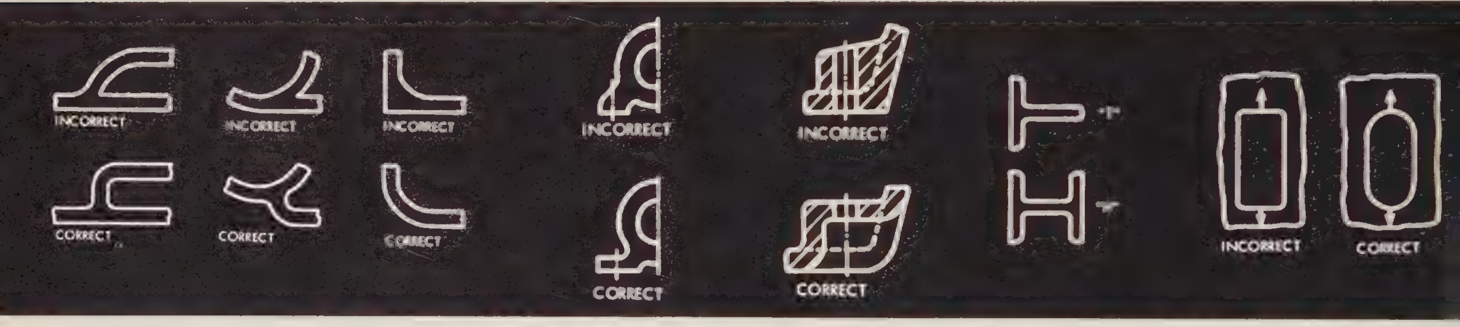


Keep material away from the neutral axis for maximum strength. The diagram shows two bars of equal cross section and length, mounted rigidly on one end. In the upper figure, the bar is mounted flat, while below, the second bar is mounted edgewise. With equal loads applied to each bar, it can be seen that the second bar has less deflection or bending.

Keep plates in tension and ribs in compression wherever possible. This diagram compares two mounting arrangements for loading plates and ribs. In the upper figure, the plate has been placed in tension and the rib in compression. This is desirable because the compressive strength of most materials far exceeds the tensile strength. The plate, which distributes the load over its entire surface, is much more effective when placed in tension, rather than compression. The rib, on the other hand, has its greatest stresses concentrated on the narrow portion of the outermost fibers. Therefore, ribs are most effective when placed in compression.

Use smoothly tapered sections to eliminate high stress concentrations. In this diagram, the lower figure shows the preferred design to avoid the severe stress concentration that occurs at the sharp corners.

Fig. 10—These are some guides to good casting design which the engineer uses along with stress analysis data.



At adjoining sections, avoid sharp corners and abrupt section changes by the use of fillets and blending radii. Here are several examples of incorrect and correct design concerning sharp corners and adjoining sections. Fillets and blending radii serve two functional purposes: (a) they reduce stress concentration in the casting, and (b) they prevent tears and draws at re-entry angles and "hot spots" which may cause local weakness and cracking as the molten metal cools. In addition, they also make corners easier to mold.

Keep casting sections as uniform in thickness as possible. Heavy casting sections present certain acute metal feeding problems while uniform sections save weight, material, machining costs, and often result in a stronger casting.

Design ribs and brackets for maximum effectiveness. "T" and "H" shaped ribbed designs have the advantage of uniform metal sections for uniform cooling. Avoid rectangular holes in ribs or webs; use oval holes with longest dimension in the direction of the stresses.

Fig. 11—Some principles of good foundry practice also guide the engineer in product design.

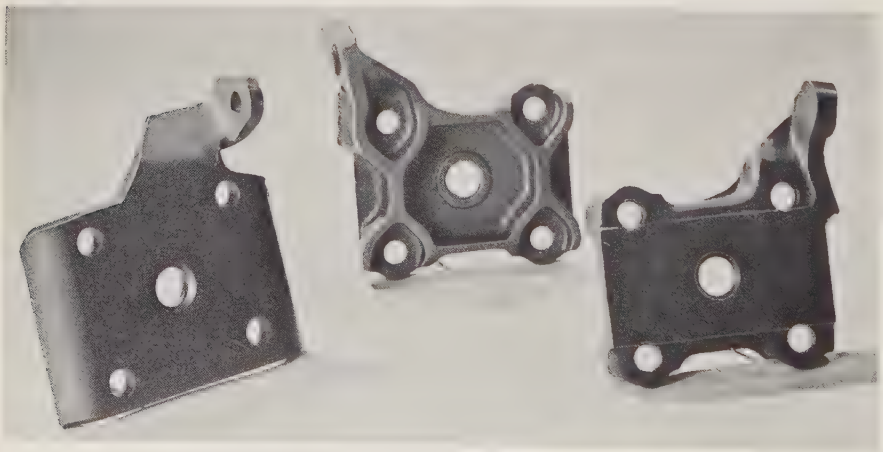


Fig. 12—The part illustrated is the rear spring clip pad used on a vehicle rear axle to clamp the spring to the axle and support the shock absorber arm. In a product improvement study, this part was changed from a stamping (left) to a casting (center and right). The stamping weighed 4.25 lb. The redesigned casting, cast from GM 86M ArmaSteel, weighed 2.9 lb, making it 32 per cent lighter. Also, the casting was 30 per cent stronger than the stamping, based on comparative tests. In the final design, the holes were cast in the part, eliminating the need for punching. It can be seen that the section thickness of the casting is uniform, and filleting is used at all adjoining sections. The thickness of the part was reduced by one-half from 7/16 in. to 7/32 in. The repositioned metal in the shock absorber *ear* reduced deflection considerably.

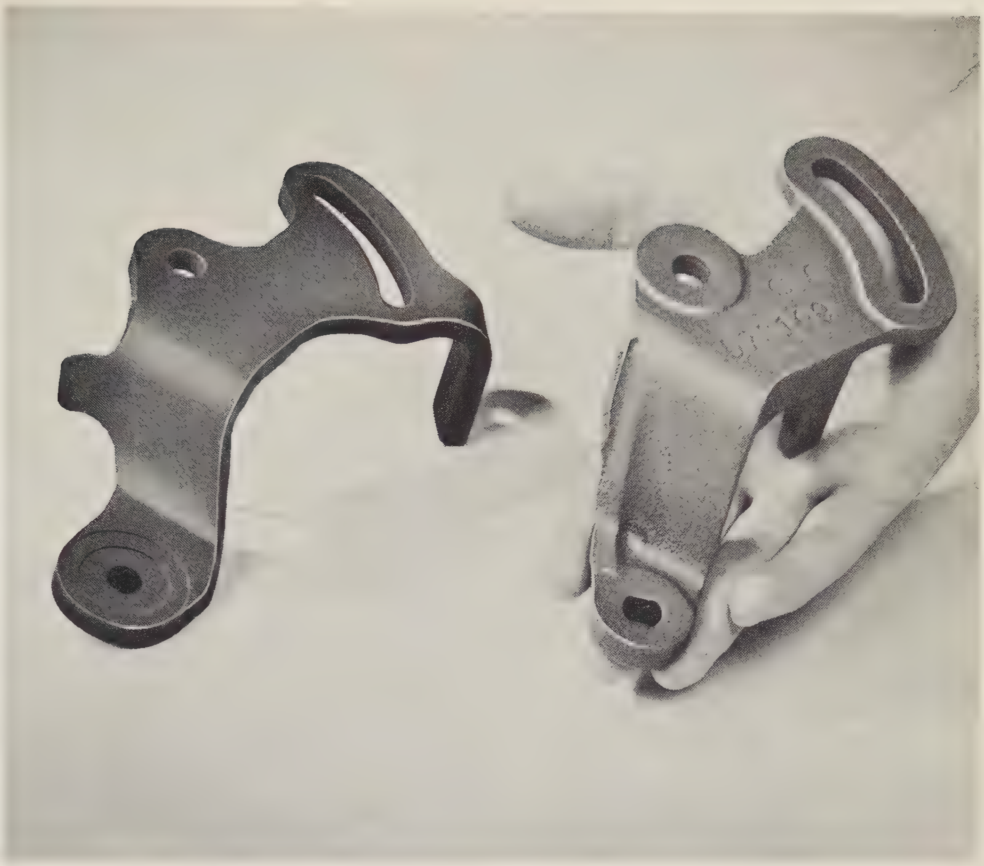


Fig. 13—Another example of converting a design from a stamping to a casting is the redesign of the pulley idler arm used for V-belt tightening adjustment on cars equipped with a compressor for air suspension. The stamping (left) was inadequate for the job due to over deflection at the rated operating load. The casting (right) was designed to minimize this deflection. The rigidity of the casting was increased by means of the gussetted rib reinforcing the part.

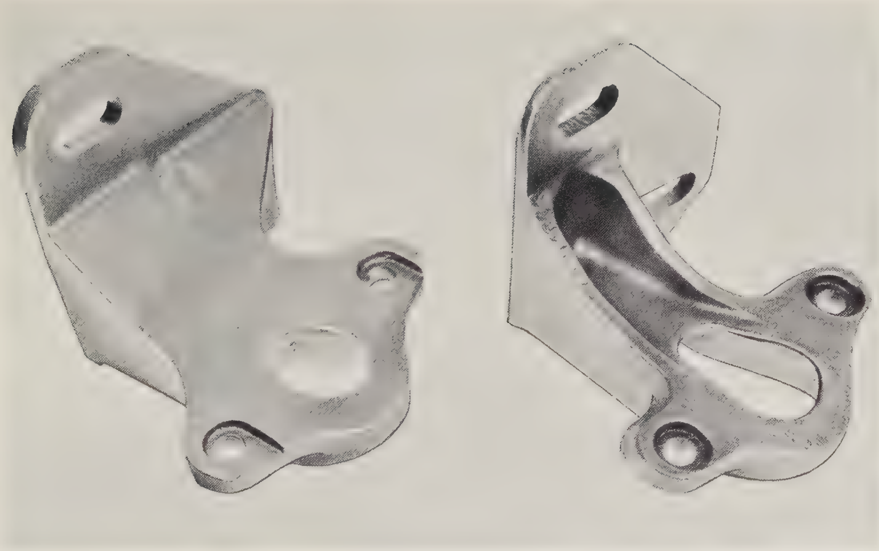


Fig. 14—The bumper support bracket shown is used on a 1959 passenger car to support the front bumper. On previous models, a strap iron assembly was used. However, for this model, a three-way bumper positioning adjustment was desired. After several design studies, it was found that a casting would serve this purpose most economically. Shown on the left is one of the first designs for this application. In order to reduce the weight and cost of this part, however, a stress analysis study was undertaken which resulted in the new design shown on the right. Here the gusset-type rib has been removed entirely, reducing the weight, and eliminating a large core.

and good foundry practice are: (a) a rear spring clip pad used on a rear axle to clamp the spring to the axle and to support the shock absorber arm (Fig. 12); (b) a pulley idler arm used for V-belt tightening adjustment on cars equipped with a compressor for air suspension (Fig. 13); and (c) a bumper support bracket used to support the front bumper of a passenger car (Fig. 14). The rear spring clip pad was changed from a metal stamping to an ArmaSteel casting with resulting savings in weight and improvement in strength for this application. Similarly, the pulley idler arm was changed from a stamping to a casting with increased rigidity as one of the results. The bumper support bracket was redesigned to reduce the weight and cost of the casting.

Conclusion

An understanding of stress analysis procedures and their application represents one way that engineers can contribute to the continuing search for improvements in casting designs and new uses for castings. Present day techniques for strain measurement provide information in the laboratory or in a test vehicle which can supplement the developmental work of the product engineer. The result can be a product that is lighter, stronger, and more economical for the user.

Dynamic Air Gaging Reduces Cycle Time in Precision Measurement

The use of air gages for measuring accurately held dimensions is not new. Air gages have been used in some General Motors Divisions since World War I. Not until World War II, however, did increased production requirements stimulate the acceptance of widespread air gaging in industry. A limiting factor in the use of air gages is the time necessary for the air pressure in the gage circuit to stabilize and arrive at a reading. GM Process Development Staff engineers recently developed a method of dynamic air gaging which allows an accurate measurement before the air pressure reaches its static value. This improvement permits a faster gaging cycle than was possible with static gaging, thus allowing a wider application of air gaging techniques in manufacturing operations.

AIR GAGING is a rapid, accurate method for measuring small distances in a range from 0.001 in. to 0.004 in. When a surface is brought within the range of an air gaging jet, it partially blocks the orifice and reduces the flow of escaping air. This causes the back pressure of air behind the jet to increase and stabilize at a new value. Since the change in pressure is proportional to the distance from the orifice to the surface, the back pressure signal may be used to indicate the measured distance either directly or on a calibrated pressure gage (Fig. 1). Also, if fed into a pressure-electric transducer, the signal may be used in sorting or feedback control.

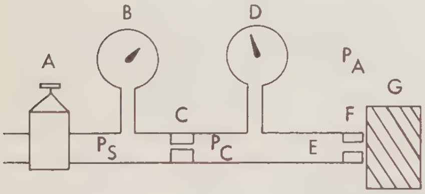


Fig. 1—The principle of air gaging is demonstrated by this simple circuit diagram. Compressed air passing through regulator A is adjusted to the desired pressure P_s (supply pressure), indicated on gage B. The air then passes through restrictor C and jet F with accompanying reductions in pressure. The chamber pressure P_c in the chamber E is indicated on gage D. When the workpiece G to be measured is not close to the jet, the air escapes readily and the chamber pressure is near that of the atmosphere P_a. As the workpiece is brought close to the jet, the escapement area is reduced greatly, decreasing the air flow and increasing the chamber pressure. If the workpiece touches the jet, air flow ceases and both gages show the same pressure. When the distance between the jet and the workpiece varies between 0.001 and 0.003 in., the pressure reading on gage D varies proportionally. This system can be used to measure distance and pressure, and gage D can be calibrated to read distance instead of pressure.

Because of the compressibility of air, the change in pressure which results from bringing a surface within the gaging range of an air jet does not happen instantaneously. The time necessary for the pressure to stabilize at a new value might be from 0.5 seconds to 2 seconds.

Although this time delay has been permissible in most measuring applications, a faster reading time would enable air gaging techniques to be applied to still more production operations.

To shorten the time required to complete a measurement by air gaging, GM Process Development Staff engineers have found that parts can be measured accurately by dynamic air gaging, that is, by measuring the pressure before it has reached its final, or static, value. This can be accomplished by taking the reading at a very definite and constant interval after the part has moved into gaging position[1].

Computer Used to Study Dynamic Air Gaging

Before the problem of dynamic air gaging was approached, a study of dynamic gaging characteristics was necessary to understand and design dynamic circuits, or systems, as easily as static circuits[1].

To set up a large number of dynamic air gaging circuits and measure the characteristics would have involved much time and effort, and would have given only empirical results. Process Development Staff engineers decided to develop a mathematical relationship between pressure, time, restrictor size, jet size, circuit volume, and distance, all of which could be fed into an analog computer to show the characteristics of the circuit. The following derivation proved sufficiently accurate for this purpose.

Mathematical Expression

The first step in applying a problem to a computer is to develop mathematical expressions for the problem. To develop an equation for a dynamic air gaging circuit, the air mass change in the back pressure chamber first had to be equated to the difference in inflow and outflow, as follows:

$$\frac{dM}{dt} = \rho\,(F_1 - F_2)$$

where

ρ = air density (lb per cu in.)
M = mass of air in chamber (lb per in.-sec^2)
F_1 = flow through restrictor (cu in. per sec)
F_2 = flow through jet (cu in. per sec).

Another value can be obtained for $\frac{dM}{dt}$ by differentiating the general gas law with respect to time t (with volume constant). If

$$PV = MRT,$$

then

$$\frac{dM}{dt} = \frac{V}{R}\left[\frac{T\left(\frac{dP}{dt}\right) - P\left(\frac{dT}{dt}\right)}{T^2}\right]$$

Therefore,

$$\frac{V}{R}\left[\frac{T\left(\frac{dP}{dt}\right) - P\left(\frac{dT}{dt}\right)}{T^2}\right] = \rho\,(F_1 - F_2).$$

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Computer analysis aids in applying "measurement by air" to more operations

It remains to write F_1, F_2, P_s, and T as functions of P, where

P_s = supply pressure (psi)
P_c = chamber pressure (psi)
P_a = atmospheric pressure (psi)
C_r = restrictor coefficient
C_j = jet coefficient.

The flow through the restrictor, F_1, can be represented by

$$F_1 = C_r \sqrt{P_s (P_s - P_c)}$$

and, for the flow through the jet,

$$F_2 = C_j \sqrt{P_c (P_c - P_a)}.$$

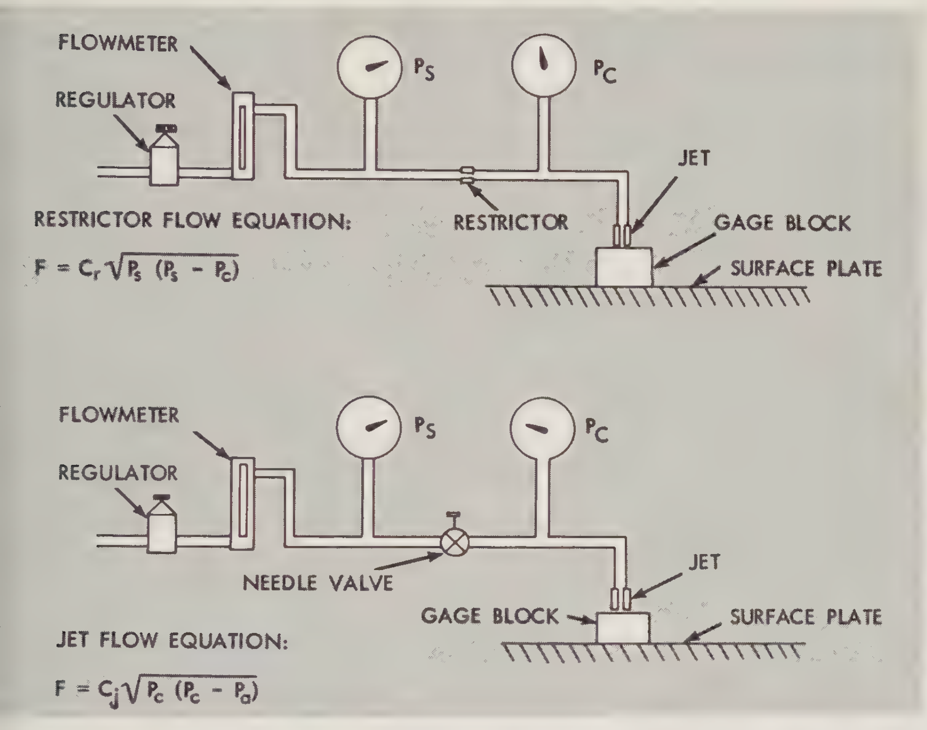

Fig. 2—Flow coefficients for restrictors and jets were determined by the use of two special air circuits. Restrictor coefficients were determined by an air circuit (top) which compared restrictor diameter with change in pressure and air flow. Applying these results to the restrictor flow equation, the restrictor coefficients were determined. In the air circuit (top), the jet diameter, distance from the jet to the workpiece, and supplied pressure P_s were held constant. Several standard size restrictors were placed in the circuit, and their coefficients determined.

Jet coefficients were determined by an air circuit (bottom) which compared jet diameter and distance from the workpiece with air flow and change in pressure. Applying these results to the jet flow equation, the jet coefficients were determined. In the air circuit (bottom), the jet diameter and distance from the workpiece were varied, while the supplied pressure (P_s) and restrictor size were held constant. The needle valve was used as a restrictor to provide easy control of restrictor diameter size.

Solving for $\frac{dP}{dt}$ gives:

$$\frac{dP}{dT} = \frac{\rho RT}{V} (F_1 - F_2) + \frac{P}{T}\left(\frac{dT}{dt}\right)$$

where

V = volume (cu in.)
P = pressure (psi)
R = gas law constant (sq in. per °R-sec^2)
T = temperature (°Rankine).

The factor T was found experimentally, and is represented by

$$T = 510 + 0.51\ (P_c).$$

The coefficients C_r and C_j were determined empirically (Fig. 2 and Table I), and the following representations were obtained:

$$C_j = d\,(\alpha + \beta\,\Delta P) + (\gamma + \delta\,\Delta P)$$
$$C_r = [D_r\,41.2 - 0.533\ (P_s - P_c)] - [0.545 - 0.0067\ (P_s - P_c)]$$

CHARACTERISTIC JET PARAMETERS

D	α	β	γ	δ
0.052	100.625	−0.355	−0.0472	−0.0004
0.073	178.23	−2.118	−0.0399	−0.0026
0.094	254.64	−3.7	0.0224	−0.0031
0.031	32.0	1.250	−0.053	0.0003

Equations for obtaining parameters α, β, γ, and δ:

1 $5a\,(\Delta P) + m\,(\Delta P)\,\Sigma D_j = \Sigma C_j\,(\Delta P)$
$a\,(\Delta P)\,\Sigma D_j + m\,\Sigma D_j^2 = \Sigma d C_j\,(\Delta P)$

2 $5\alpha + \beta\Sigma\,(\Delta P) = \Sigma m\,(\Delta P)$
$\alpha\Sigma\,(\Delta P) + \beta\Sigma\,(\Delta P)^2 = \Sigma \Delta P m\,(\Delta P)$

3 $5\gamma + \delta\Sigma\,(\Delta P) = \Sigma a\,(\Delta P)$
$\gamma\Sigma\,(\Delta P) + \delta\Sigma\,(\Delta P)^2 = \Sigma\,(\Delta P)\,a\,(\Delta P)$

where

ΔP = change in pressure (psi)
D_j = jet diameter (in.)
C_j = jet coefficient
d = distance of workpiece from jet (in.)
a = function of parameter
m = function of parameter.

Table I—The characteristic jet parameters α, β, γ, and δ, are shown in this table for specific jet diameters (listed in column *D*). These parameters were obtained from the equations shown. In the equations, which were set up from known relationships between jet characteristics, the functions *a* and *m* were developed to simplify the formulation of the equations.

where

d = distance of work from jet (in.)
D_r = restrictor diameter (in.)
D_j = jet diameter (in.)
$\alpha, \beta, \gamma, \delta$ = characteristic parameters of jet.

The complete set of equations is as follows:

$$\frac{dP}{dt} = \frac{\rho RT}{V}\left[C_r \sqrt{P_s (P_s - P_c)} - C_j \sqrt{P_c (P_c - P_a)}\right] + \frac{P}{T}\left(\frac{dT}{dt}\right) \quad (1)$$

$$C_r = D_r [41.2 - 0.533\ (P_s - P_c)] - [0.545 - 0.0067\ (P_s - P_c)] \quad (2)$$

$$C_j = D_j [41.2 - 0.533\ (P_c - P_a)] - [0.545 - 0.0067\ (P_c - P_a)] \text{ when } t < 0^* \quad (3)$$

$$C_j = d\,[\alpha + \beta\ (P_c - P_a)] + [\gamma + \delta\ (P_c - P_a)] \text{ when } t > 0 \quad (4)$$

$$T = 510 + 0.51\ (P_c)$$

$$P_s = P_{so}\,[1 - 0.1\ (C_r \sqrt{P_s (P_s - P_c)}]$$

where

P_{so} = initial supply pressure when $t < 0$.

Machine Equations

Once the mathematical relationship between the various factors affecting the dynamic air gaging circuit were known,

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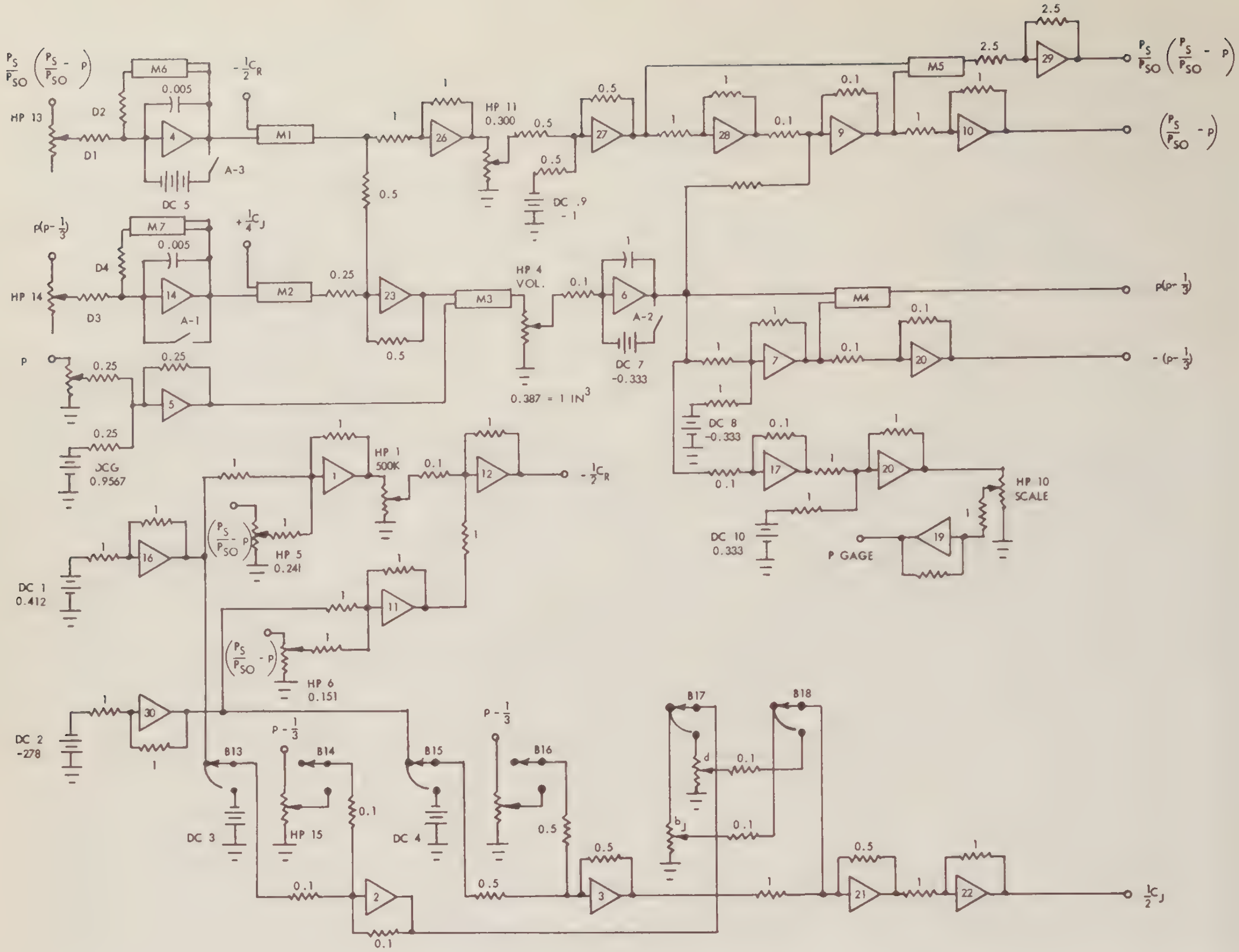
*When no part is in front of the jet it acts as a restrictor and follows a similar formula.



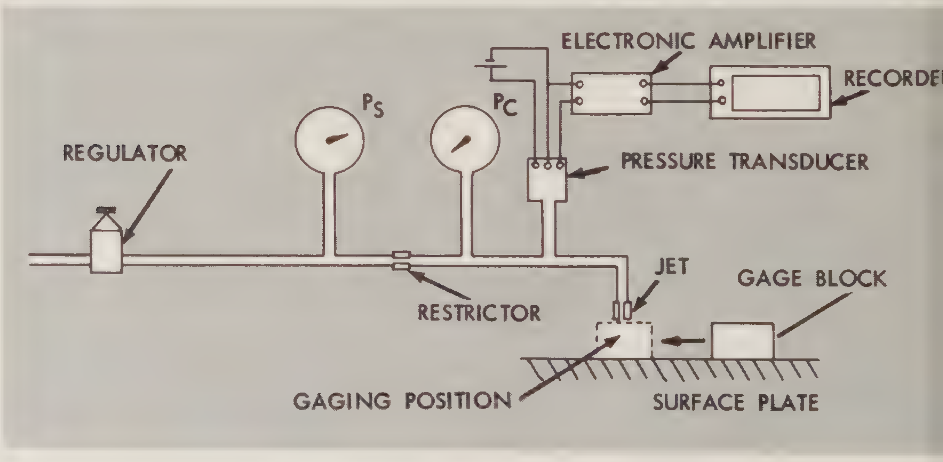


Fig. 3—The analog computer schematic diagram (top) shows the electronic circuitry of the computer used to investigate air gaging circuits. The air gaging circuit (right) shows the dynamic circuit used to check the computer curves.

equations for the analog computer were developed. In this problem, the computer was prepared to relate time against pressure to show the characteristics of the circuit being examined.

In the development of the machine equations, the conversion factors were as follows:

$$p = \frac{P_c}{P_{so}}$$

$$P_a = u\,(P_{so})$$

u = a constant

$$\Theta = \frac{T}{533}$$

$$R = 640$$

$$\rho = 4.6\ (10^{-5})$$

$$\lambda = 10t$$

$$10\,\frac{d}{dt} = \frac{d}{dt}.$$

Dropping the last term

$$\frac{P}{T}\left(\frac{dT}{dt}\right)$$

from equation (1) as an infinitesimal, and substituting the conversion factors

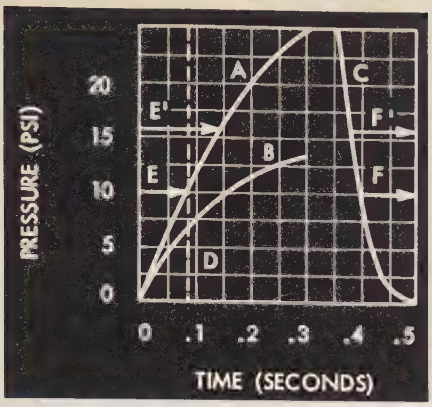


Fig. 4—The typical graph of the characteristics of an air gaging circuit, as drawn by the computer, contains three curves. Curves *A* and *B* show the pressure rise with time when a workpiece is at the minimum and maximum extremes of the gaging range of the circuit. Curve *C* shows the rate of pressure decay when the workpiece is removed from the gage.

The vertical dotted line *D* represents the time interval for the circuit to reach a sensitivity of 0.2. This sensitivity line is the gaging time necessary for the circuit to reach 0.20 psi per 0.0001 in., and is used to evaluate the gaging speed of the circuit. The line marks the end of the minimum gaging time and the start of the recovery curve for a minimum gaging cycle. The 0.2 sensitivity was chosen arbitrarily because most pneumatic and air-electric relays used for sorting and control are sensitive to a 0.2 psi pressure change.

The horizontal vectors *E* and *F* represent the minimum gaging and decay times for the circuit, respectively. When added together they give the minimum gaging cycle time (in this case 0.09 + 0.1 = 0.19 sec). Similarly, vectors *E'* and *F'* give the cycle time when the workpiece is gaged at 0.15 sec (0.15 + 0.12 = 0.27 sec). To obtain any gaging cycle time, the gaging time is selected, and vector *E* is drawn. At the same pressure, vector *F* is drawn, and the two vectors are added.

into equations (1), (2), (3), and (4), the following machine equations were developed:

$$10\,P_{so}\,\frac{dP_c}{d\lambda} = \frac{4.6\,(10^{-5})\,(640)\,(533\,\Theta)}{V}\left[C_r\,(P_{so})\sqrt{\frac{P_s}{P_{so}}\left(\frac{P_s}{P_{so}}-p\right)} - C_j\,(P_{so})\sqrt{p-(p-u)}\right]$$

$$\frac{dP_c}{d\lambda} = 1.57\Theta\left[C_r\sqrt{\frac{P_s}{P_{so}}\left(\frac{P_s}{P_{so}}-p\right)} - C_j\sqrt{p-(p-u)}\right]$$

$$C_r = D_r\left[41.2 - 0.533\,(P_{so})\left(\frac{P_s}{P_{so}}-p\right)\right] - \left[0.545 - 0.0067\,(P_{so})\left(\frac{P_s}{P_{so}}-p\right)\right]$$

$$C_j = D_j\,[41.2 - 0.533\,(P_{so})\,(p-u)] - [0.545 - 0.0067\,(P_{so})\,(p-u)] \quad \text{when } t < 0$$

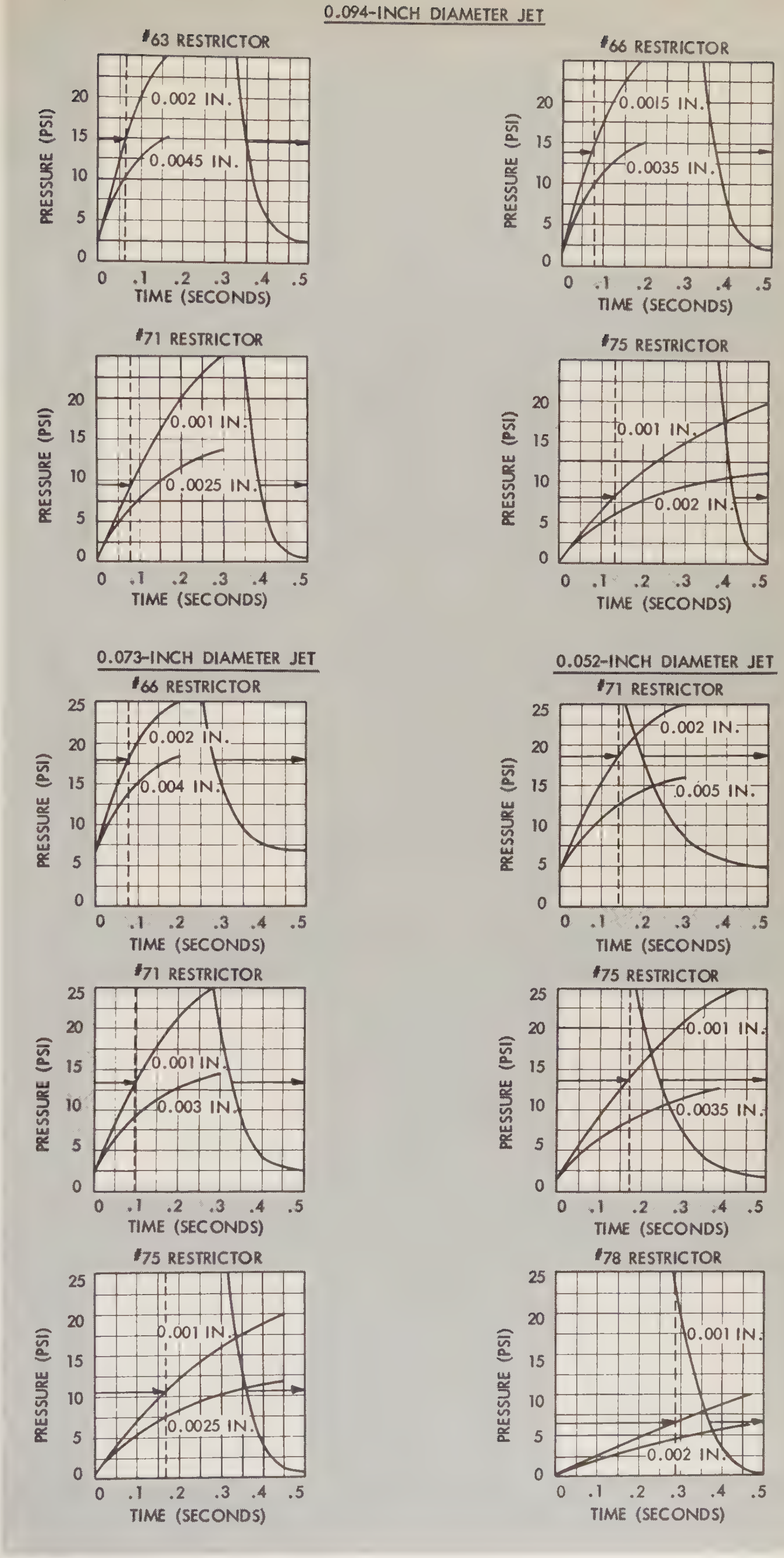


Fig. 5—The characteristic curves of the dynamic air gaging circuits shown here were drawn by the analog computer. These particular jet and restrictor sizes are the ones most commonly used in air gaging. The restrictor size number refers to the specification of the drill used to bore the restrictor orifice.

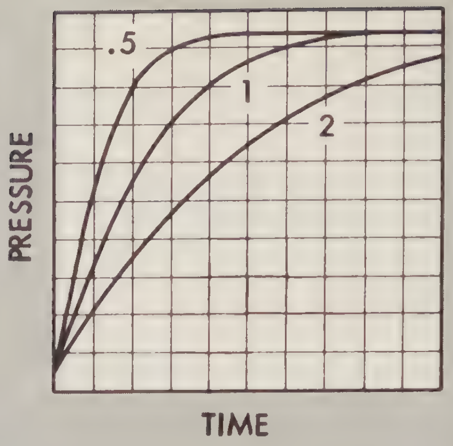


Fig. 6—The volume in the air circuit is an important factor in air gaging, since it is directly proportional to the gaging speed. This graph indicates the variation in circuit gaging speed with air volumes of 0.5, 1, and 2 cu in.

$$C_j = -d\,[\alpha + \beta\, P_{so}\,(p - u)] + [\gamma + \delta\, P_{so}\,(p - u)] \text{ when } t > 0$$

$$\Theta = 0.043p + 0.9567.$$

Curves Drawn by Computer Show Gaging Characteristics

A computer circuit (Fig. 3) was arranged to correspond with the developed equations. Using this computer circuit, a series of characteristic curves were drawn which fully described the behavior of the dynamic air gaging circuits.

A typical graph contained three curves (Fig. 4). The coordinates of these curves represented pressure (psi) and time (seconds). They showed the pressure rise with time when a part was at the maximum and minimum extremes of the gaging range of the given circuit, and the rate of pressure decay when the part was removed from the gage. Curves were drawn for the most commonly used air gaging circuits (Fig. 5).

A study of the curves shows that a dynamic air gaging cycle is composed of two parts, a *gaging time* and a *recovery time*. Gaging time is the time during which the pressure builds up because of the workpiece in front of the orifice. These two phases comprise the *gaging cycle*, which is the time necessary to place a part before the orifice, remove it, and replace it with another part.

As the jet diameter is increased, both parts of the gaging cycle are shortened. For a given jet size, as the restrictor diameter is increased, the minimum gaging time is shortened, but the recovery time is lengthened, and the total gaging cycle time remains constant.

It is very important that the pressure be allowed to stabilize after a gaged part is removed and before the next part is placed before the orifice. If the pressure is not allowed to stabilize, the gaging pressure curve rises from a false datum and gives a high reading. This imposes certain limitations on dynamic air gaging. For example, cylindrical parts of uniform cross section cannot be pushed out of the gage head by the next part because the circuit does not have time to stabilize. Where sufficient time is available for stabilization between measurements, the actual gaging time may be as short as 0.05 seconds.

The volume of the circuit has a great effect on the gaging speed (Fig. 6). The gaging speed is directly proportional to the volume. In a well designed conventional air gaging circuit, one cubic inch is about as small a volume as can be achieved.

Curves Aid in Selection of Circuits

To design a dynamic air gaging circuit for a specific gaging problem, the gaging time, total gaging cycle, and gaging range (maximum and minimum dimensions) must be known. After a study of the problem has revealed these factors, reference to the characteristic curves shows the circuits which can handle the problem. The satisfactory circuit with the smallest jet size is usually chosen, since it uses the least amount of air.

As an example, assume that in a certain operation it is desired to measure a dimension of 1.000 in. plus or minus 0.0005 in. at a rate of 10,000 parts per hour, with a maximum gaging period of 0.2 seconds. Therefore, the air gaging circuit must meet the following requirements:

- gaging range = 0.001 in.
- gaging time = 0.2 sec maximum
- gaging cycle = 0.36 sec.

Referring to the characteristic curves (Fig. 5), at least three satisfactory circuits can be found:

- One circuit, using a 0.073-in. jet diameter and No. 75 restrictor, gives a cycle time of 0.345 sec, a linear range of 0.0014 in., and a sensitivity of 0.23
- The second circuit, using a 0.094-in. jet diameter and No. 75 restrictor, gives a cycle time of 0.29 sec, a linear range of 0.0012 in., and a sensitivity of 0.35
- The third circuit, using a 0.094-in. jet diameter and a No. 71 restrictor, gives a cycle time of 0.33 sec, a linear range of 0.0017 in., and a sensitivity of 0.53.

The linear range values are obtained from *Air Jet Characteristics*, published by the General Motors Process Development Staff.

From the above analysis, the 0.073—No. 75 air circuit meets the requirements with the lowest air consumption of the three.

Conclusion

Using an analog computer to simulate dynamic air gaging, Process Development Staff engineers proved the feasibility of dynamic air gaging. Gaging rates of 15,000 measurements per hour are possible with dynamic gaging, and the actual gaging time may be as short as 0.05 seconds.

It was found that sufficient time must be allowed between gaging operations for the pressure to stabilize, and it was discovered that gaging time is proportional to the circuit air volume. A valuable property of dynamic air gaging is that the fastest circuits have the greatest sensitivity and linear range.

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Firebird III: A New Concept in Automotive Engineering

STYLED and engineered for the Space Age might be an apt description for Firebird III, latest in a series of GM experimental gas turbine powered vehicles (Fig. 1). A two-passenger car, the Firebird III has been designed to permit the study and evaluation of advanced automotive engineering concepts under actual operating conditions. Strictly experimental and not intended for production, the car has undergone extensive tests at the GM Technical Center; GM Proving Ground, Mesa, Arizona; and on public highways. Developed as a joint project by the GM Research Laboratories and the Styling Staff, with various component contributions made by the Engineering Staff and 10 GM Divisions, the Firebird III was first displayed to the public at the 1958 GM Motorama.

Many new concepts in automotive engineering are embodied in Firebird III, chief of which is the car control system—a significant step in a long-range investigation by the Research Laboratories to find the simplest, safest, and most efficient method for controlling a vehicle. The complete system consists of three interrelated sub-systems: Unicontrol, a means of controlling all steering, braking, and accelerating operations with a single lever; Autoguide, a means of automatic steering on the highway; and Cruise-control, a means of maintaining a pre-set speed.

Unicontrol combines all accelerating, braking, and steering operations into a single swivel stick topped by a 1½-lb handle (Fig. 2). The stick is moved forward for acceleration, backward for braking, and left or right for steering. The stick produces any possible combination of driving operations by positioning either or both of two electrical potentiometers attached to its base. One potentiometer feeds a voltage to an electronic analog computer in the steering system, and the other feeds a voltage to an electronic system which controls the throttle and brakes.

The principle of operation is essentially the same for each control channel. For example, assume that the steering system computer receives a signal from the hand control potentiometer, another from the tachometer generator showing car velocity, and a third from an indicator showing

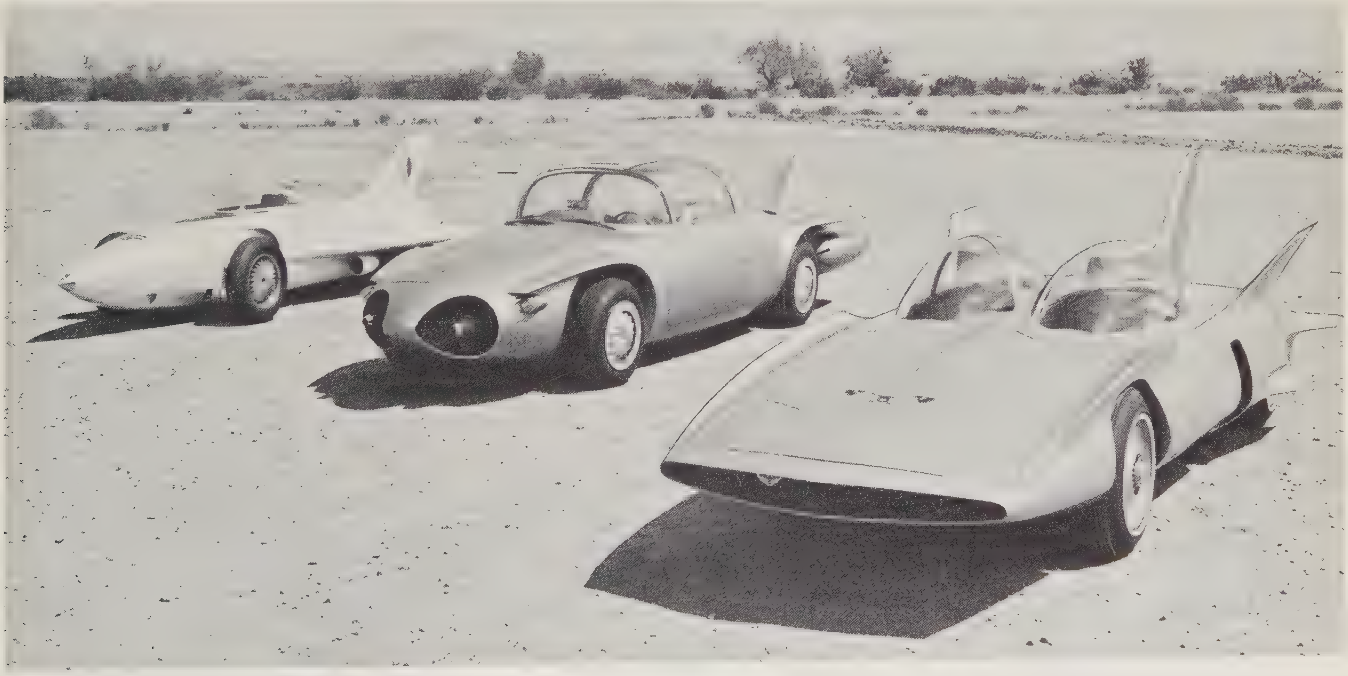

Fig. 1—To date, three experimental gas turbine powered vehicles have been developed by GM to test and evaluate new design ideas in various areas of automotive engineering. Shown here, from left to right, are Firebirds I, II, and III. Firebird I was introduced in 1953 solely to study the possibilities of the gas turbine as an automotive power plant and was not designed for highway use. Firebird II, announced in 1955, was designed for highway travel and represented progress made in gas turbine engine design for automotive application and also new concepts in passenger comfort. Firebird III, which has a Fiberglas body, represents a progress report on advanced engineering in the gas turbine-automotive application area and, in addition, dramatizes the results achieved by GM engineers, stylists, and scientists in the application of such engineering tools as electronics, analog computers, and transistors to automatic car control and increased passenger comfort and safety. Firebird III was first displayed to the public at the 1958 GM Motorama.

front wheel position. This computer takes these three signals and computes a stabilizing signal which is flashed to an electrohydraulic control valve positioned on the front wheel servomechanism. This control valve then regulates a flow of high pressure oil to a power piston which operates to turn the front wheels. The throttle and brakes are controlled in a similar manner by individual computers and servomechanisms.

The steering system is designed so that the front wheel angle for a given stick displacement varies inversely as the square of the speed. At low speeds and during parking maneuvers, a small stick movement produces a relatively large front wheel displacement. At higher speeds the same movement produces a much smaller wheel displacement. No matter what the speed, usable stick movement remains constant and will not throw the car into a skid on dry pavement.

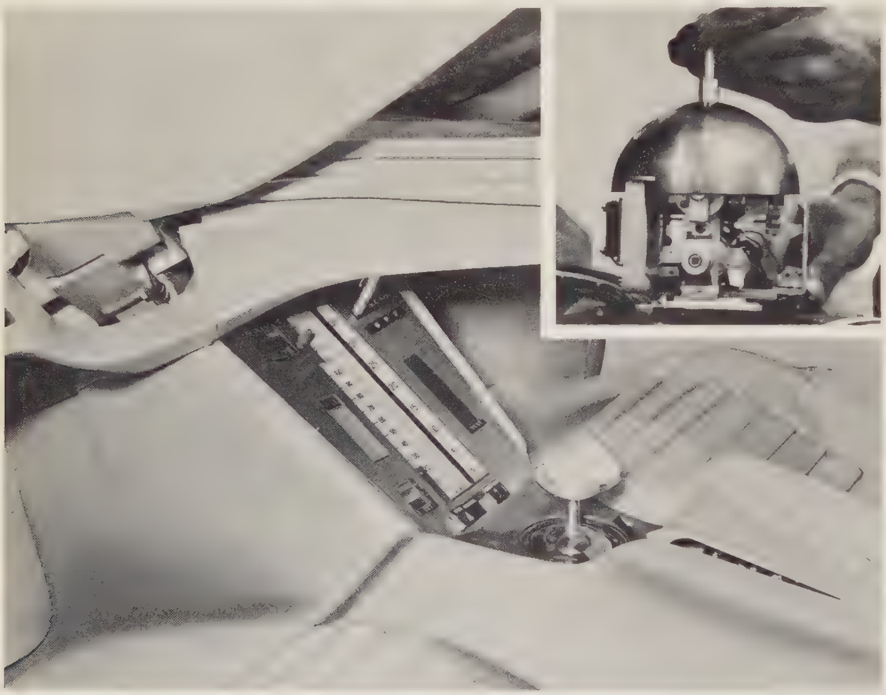

Fig. 2—All steering, accelerating, and braking operations for the Firebird III are performed by means of a single control stick, called *Unicontrol*, located in the center of the passenger compartment so as to be easily operated by either passenger. Forward movement of the stick causes acceleration; backward movement causes braking; and movement to the left or right steers the car accordingly. Movement of the stick positions potentiometers, attached to the base of the stick (inset), which feed electrical inputs to analog computers. These computers determine proper voltages for electrohydraulic servomechanisms located on the front wheels, the throttle, and the brakes. A feature of the instrument panel is a warning section which has 13 tell-tale lights, each designating a specific primary operating component of the car. A malfunction in one of these components is indicated to the driver by means of a flashing light.

The heavy handle atop the Unicontrol stick provides a built-in skid control. Because the stick is top heavy it acts like an inverted pendulum, and is quite sensitive to lateral acceleration. If the car should skid, the inertia of the handle tilts the stick and causes the wheels to turn in the proper direction for recovery. Handle inertia also serves to supply car motion feedback to the driver, thereby giving a feel of the road, even on extremely slippery surfaces.

The Unicontrol stick has another safety feature. Because the stick is spring loaded and damped, it always returns to a neutral position without overshooting. Consequently, the car always rights itself rapidly and without oscillation when the stick is released. In addition, the rigid positioning of the front wheels by the servomechanism provides a high degree of course stability. The Unicontrol stick handle also serves as a transmission selector and includes a button for actuating a grade retarder unit on the automatic transmission. For Reverse, the handle is rotated 20° to the right or left. For Park, the handle is rotated 80° to the left or right. In each case, the stick remains completely mobile.

Autoguide represents a further refinement to an automatic steering system control demonstrated by GM engineers earlier this year[1]. The system works on the principle that a conductor carrying alternating current radiates an electromagnetic field along its entire length and that this field can be used for automatic guidance purposes. To effect automatic steering, two cylindrical pickup coils, connected in opposition to each other, swing down from the forward belly of Firebird III into position above a cable embedded in the road surface. The alternating field of the cable induces a voltage in both coils the value of which depends on the proximity of the coils to the cable. The pickup coils feed corrective steering signals to the steering system computer whenever the car deviates from its straight-ahead course. When Autoguide is in operation, the Unicontrol stick retains its control over acceleration and braking.

Cruisecontrol is part of the throttle control setting and, when in operation, assumes this function from the Unicontrol stick. Cruisecontrol automatically maintains whatever road speed is set on a dial. The pre-set signal is sent to the throttle computer along with a signal representing car speed from a tachometer generator, coupled to the automatic transmission. The computer uses this input information, along with a signal indicating throttle position, to determine the correcting signal for the electrohydraulic control valve on the throttle servomechanism. The throttle setting, therefore, is increased when car speed falls below the Cruisecontrol setting and decreased when car speed exceeds the setting.

Both Unicontrol and Cruisecontrol can be used with present day highways. Autoguide, however, is designed to operate in accordance with command information from a suitably equipped automatic road system. Supplementing Autoguide with Cruisecontrol gives a combination approaching that required for complete automatic highway control.

In addition to the car control system, another primary engineering feature of Firebird III is the Whirlfire GT-305 regenerative gas turbine engine (Fig. 3), which is more compact, more powerful, and more economical in operation than the GT-304 used in Firebird II[2] or the

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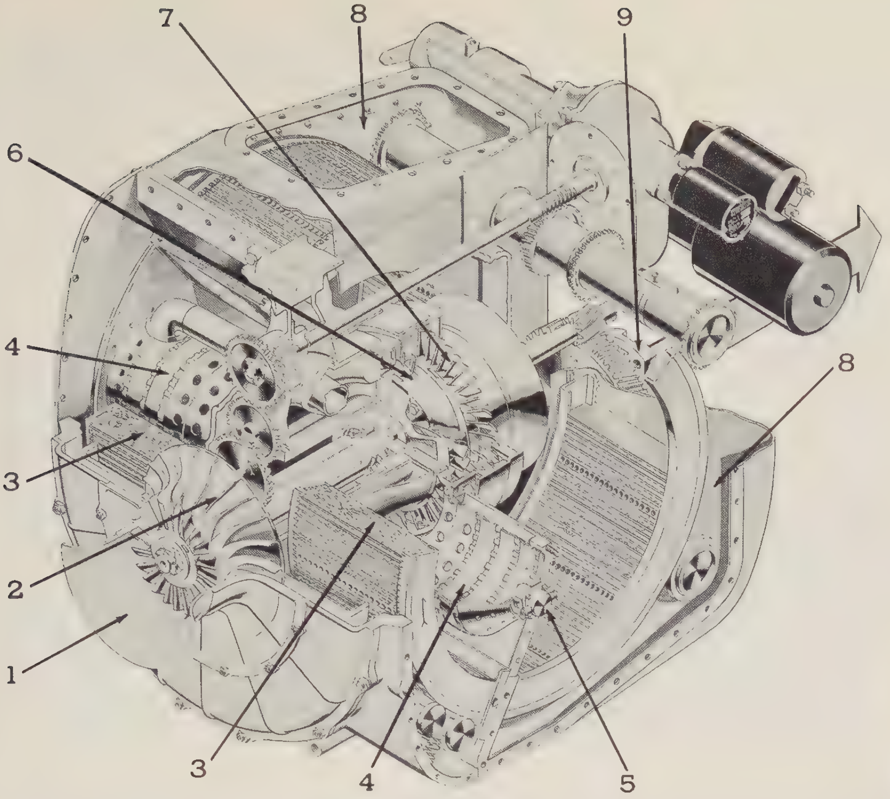
[1] "GM Engineers Develop Electro-Magnetic Guidance System to Steer Vehicles Automatically," General Motors Engineering Journal, Vol. 5, No. 2 (April-May-June 1958), pp. 60-1.

[2] "Firebird II: New Gas Turbine Powered 'Dream Car' Designed for Highway Use," General Motors Engineering Journal, Vol. 3, No. 2 (March-April 1956), pp. 43-5.



modified GT-304 used in an experimental gas turbine powered truck[3]. The GT-305 is 32.1 in. in length and 23.8 in. in overall height. The engine weighs 600 lb and generates 225 hp at a gasifier speed of 33,000 rpm. This represents a 25 per cent reduction in weight and a 10 per cent increase in power compared to the GT-304 engine.

The GT-305 engine has independent gasifier and power sections and a regenerator which utilizes exhaust heat to conserve fuel. The gasifier section is simply a shaft with a centrifugal com-

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[3] "GM Engineers Develop Experimental Gas Turbine Powered Truck," General Motors Engineering Journal, Vol. 4, No. 2 (April-May-June 1957), p. 21.

Fig. 3—The engineering improvements in gas turbine power for automotive application are emphasized in Firebird III's GT-305 Whirlfire regenerative gas turbine engine, shown above in a cutaway view. The GT-305 is of smaller size and simpler construction, compared to the GT-304 engine used in Firebird II. Design refinements have made possible a 25 per cent reduction in both fuel consumption and weight and a 10 per cent increase in power. Air enters the inlet *1* and is compressed by the centrifugal compressor *2* to over three atmospheres. The compressed air then absorbs exhaust heat while passing through two rotating regenerators *3*. The heated air then enters the combustors *4* where fuel injection nozzles *5* add fuel for combustion. The combustion gases pass through turbine vanes to drive the gasifier turbine *6*. The gases leaving the gasifier turbine then drive the power turbine *7*. The power turbine is not mechanically connected to the gasifier shaft. The hot exhaust gases are then cooled as they pass through the rotating regenerators. The relatively cool exhaust gas (500°F) is then directed out through exhaust ports *8*. The power turbine is connected to the engine drive shaft *9* through a single-stage helical reduction gear.

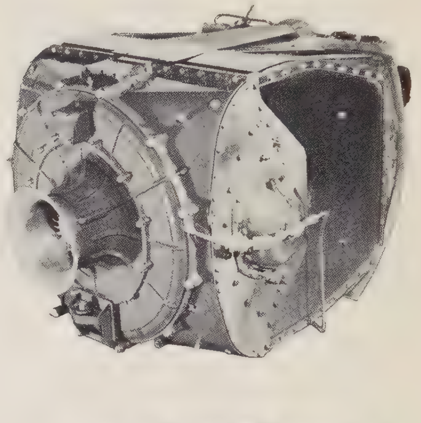
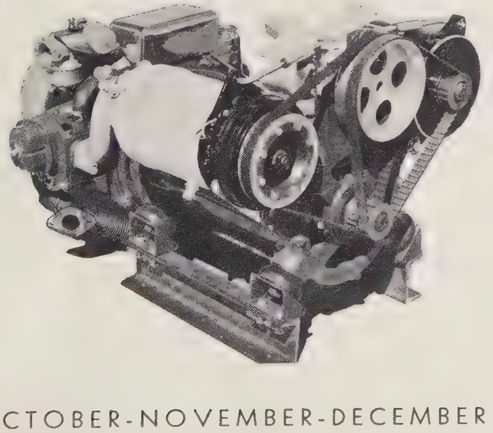
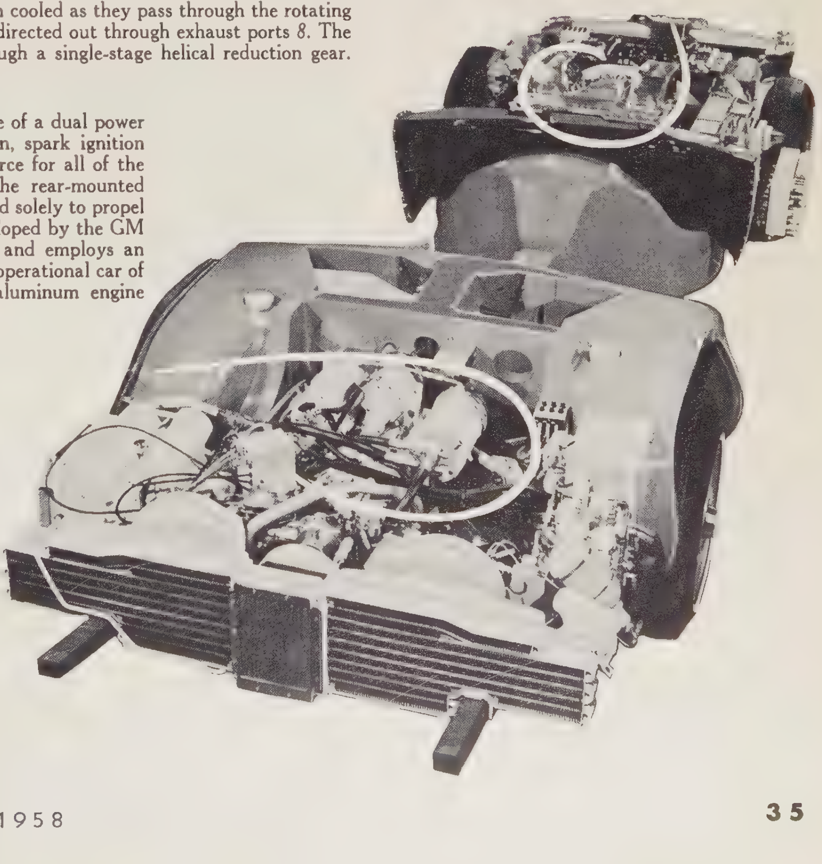


Fig. 4—A unique feature of the Firebird III chassis is the use of a dual power plant. A 10-hp, two-cylinder, constant speed, opposed piston, spark ignition engine mounted at the front (inset) serves as the power source for all of the car's electrically and hydraulically operated accessories. The rear-mounted GT-305 Whirlfire regenerative gas turbine engine (inset) is used solely to propel the car. The 10-hp accessory drive engine, designed and developed by the GM Engineering Staff, is built of wear-resistant cast aluminum and employs an integral head design. Its use marks the first application in an operational car of a liquid cooled, high compression (11 to 1), high-output aluminum engine without liners.





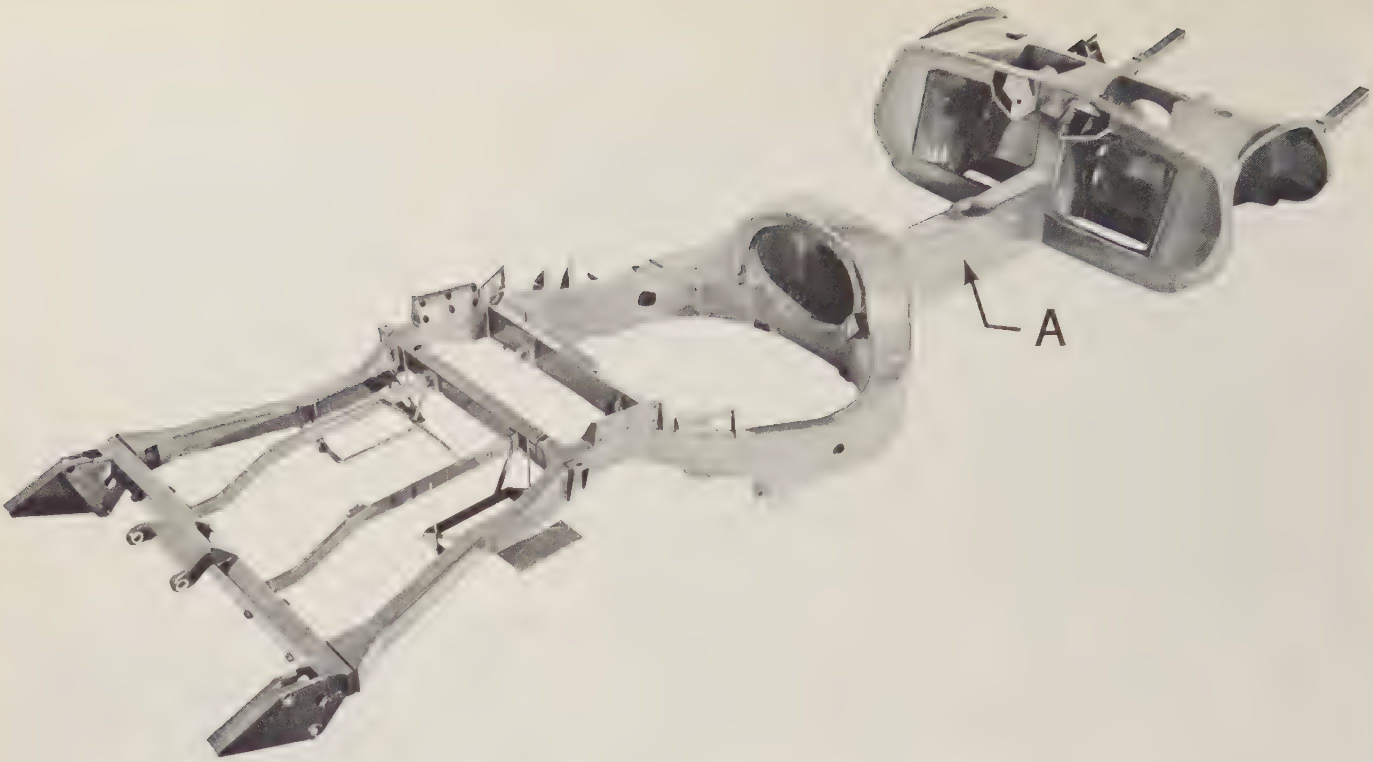


Fig. 5—The frame for the Firebird III has a short longitudinal center spine section *A* which serves as the principal structural member and provides the required bending and torsional stiffness through the passenger compartment. This spine section also serves as a conduit for electrical and hydraulic lines running to the rear and as a housing for the main car operating controls. The front frame section, welded to the spine section, includes the two side rails, cowl, and integrated front wheel housings which incorporate rigid seats for suspension displacement cylinders. The rear end of the spine section blends into a housing for the air intake of the gas turbine engine. A series of baffle plates within the housing serve as the engine silencer. (Air inlets to the engine are located on either side of the car, just behind the doors.) The rear side rails are an extension of the air intake housing. The frame and all supporting stringers for the body are welded into one unit with provision being made for mounting separate Fiberglas body panels.

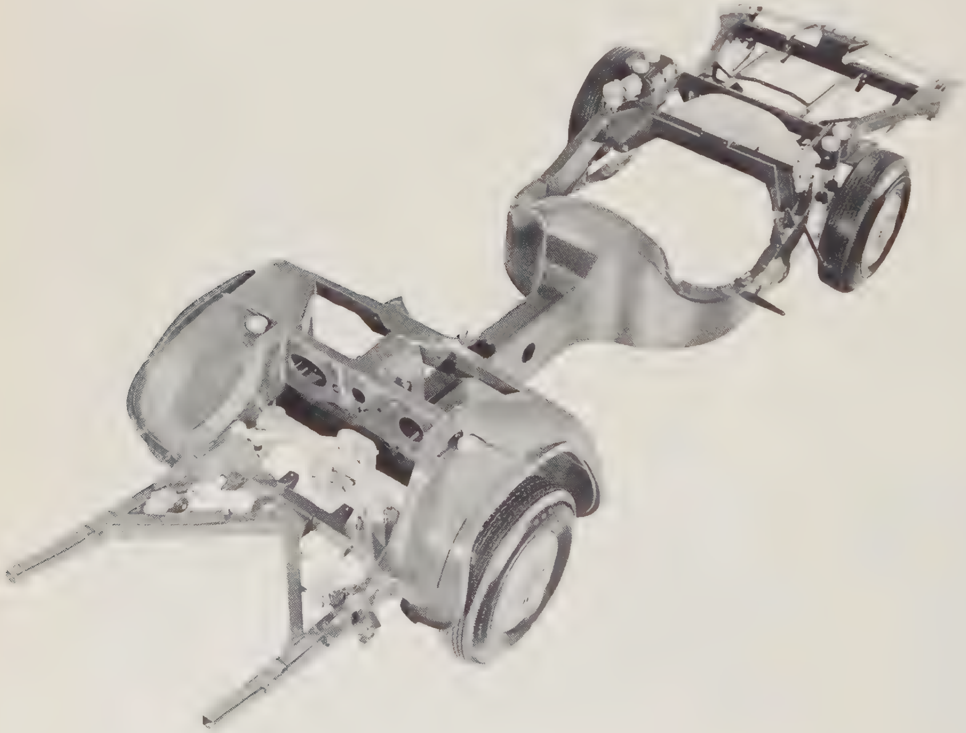


Fig. 6—Steering gear components of the Firebird III are attached directly to a solid front axle. Since these components are independent of the frame, steering geometry is not affected by relative movement between the frame and axle. The GT-305 engine, modified Hydra-Matic automatic transmission, and the differential are coupled together and mounted as a single unit behind the passenger compartment. Three point mounting in rubber is used. The power train arrangement eliminates the usual propeller shaft and universal joints. A De Dion rear axle arrangement is used. The suspension system embodies an interconnected front and rear air-oil spring arrangement. When the front wheels of the car pass over a bump, any vertical force acting on the front of the car is also applied simultaneously to the rear through the interconnected air-oil springs. The same interaction takes place when the rear wheels pass over the bump. The net result is elimination of pitching motions and provision for a softer ride and greater roll stiffness.

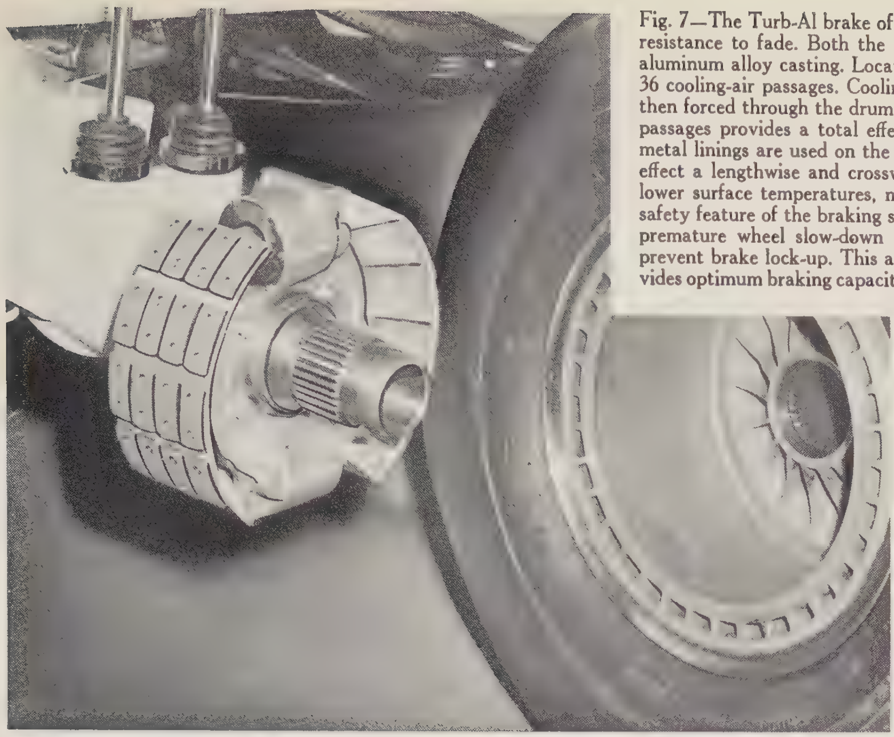


Fig. 7—The Turb-Al brake of the Firebird III provides consistent braking with high resistance to fade. Both the wheel and brake drum are combined into an integral aluminum alloy casting. Located between the drum and wheel sections is a series of 36 cooling-air passages. Cooling air enters through 18 main inlets at the hub and is then forced through the drum and ejected at the wheel rim. The large number of air passages provides a total effective cooling surface area of over five sq ft. Sintered metal linings are used on the cast aluminum brake shoes. Each shoe has 16 pads to effect a lengthwise and crosswise groove pattern. This configuration contributes to lower surface temperatures, more uniform braking, and longer wear. An important safety feature of the braking system is an anti-skid device. A sensing element detects premature wheel slow-down automatically and maintains proper line pressure to prevent brake lock-up. This almost instantaneous action prevents skidding and provides optimum braking capacity and steering control even under panic stop conditions.

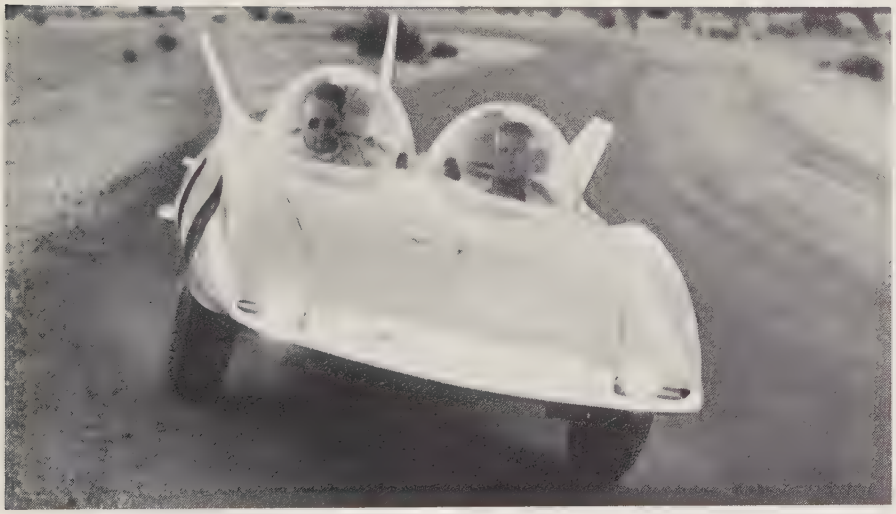


Fig. 8—The lighting system of the Firebird III is unique in several respects. The lower beam headlamp is a fluorescent-type lamp which, in conjunction with a specially designed parabolic reflector, is mounted in a recess at the front of the car. Sealed beam, aircraft-type lamps of 100-watt rating are used for the upper beam headlamps. An automatic light control turns the headlamps on or off depending on ambient light intensity. The controlling signal is provided by three, light-sensitive pickups mounted externally on the body. This system serves as an On-Off control for the lower beam headlamp, upper beam headlamps, parking, and tail lamps.

pressor on the front end and a single-stage, axial flow turbine at the other end. The compressor operates at a pressure ratio of 3.5 to 1 and the gasifier turbine rotates at a full-power speed of 33,000 rpm. The power turbine, which rotates at 27,000 rpm, is also of the single-stage, axial flow design, and is not mechanically connected to the gasifier shaft. It receives its rotative motion from the impinging stream of hot gases leaving the gasifier turbine and is connected to the engine drive shaft through a single-stage helical reduction gear having a 6.9 to 1 ratio. The drive shaft transmits power to a modified Hydra-Matic automatic transmission and from there to the differential and rear wheels.

A 25 per cent improvement in fuel economy has been achieved in the GT-305, when compared to the GT-304, through various design refinements and simplification. One component contributing to improved fuel economy is the regenerator, which rotates at approximately 20 to 30 rpm. The regenerator rotates first through the hot exhaust gases and then through the relatively cool compressor discharge air. Thus, heat is carried from the exhaust gas to the compressed air. As a result, less fuel is burned in bringing the hot air to its operating temperature of 1,650°F at the gasifier turbine inlet. Also, the exhaust gas temperature is reduced to about 500°F, or roughly half that of a conventional automotive piston engine. The regenerator recovers from 85 to 90 per cent of the recoverable heat in the exhaust compared to about 80 per cent for the regenerator of the GT-304 engine.

Another factor contributing to improved fuel economy is the provision for two idle speeds. A low idle speed of about 12,000 rpm is used for long idling periods to conserve fuel. A higher idle speed of approximately 17,000 rpm is used for normal stop-start driving to provide improved acceleration performance. Also contributing to improved fuel economy is the use of two, can-type combustion chambers rather than the four chambers used in the GT-304 engine. This design not only saves space but the higher flow rate of each fuel injection nozzle provides a better spray pattern for more efficient combustion.

Other engineering features of Firebird III include an auxiliary 10-hp engine used to provide power for all electrically and hydraulically operated accessories (Fig. 4); a unique frame design (Fig. 5); an air-oil suspension system (Fig. 6); a brake design, *Turb-Al*, in which the wheel and brake drum are combined into a single aluminum alloy casting (Fig. 7); and a headlamp lighting system (Fig. 8) which turns on automatically when light intensity reaches a specific level.

The Filing and Maintenance of Patents in Foreign Countries

By FRED E. JONES
Patent Section
Detroit Office

An outline of some of the problems encountered

PREVIOUS issues of the GENERAL MOTORS ENGINEERING JOURNAL have included discussions of inventions and patent procedures which applied primarily to applications in the United States. Applications for patents in other countries of the world often are filed by American inventors, and these foreign applications are governed by a variety of practices. It may be helpful, therefore, to examine some of the steps in obtaining patents in foreign countries on inventions which are made in the United States and which are covered by applications filed with the United States Patent Office.

Under the provisions of the International Convention for the Protection of Industrial Property, it is possible for an applicant, who has first filed in a country which is a member of the Convention, to file corresponding applications in other Convention countries and to become entitled in such countries to the priority of his original filing date if he files therein within 12 months from such original filing date. The advantage of this is that it gives applicants more time to decide whether to file in foreign countries without risking invalidation of corresponding foreign patents by intervening publication or use, or by intervening similar applications, of others.

The United States and most of the principal foreign countries are members of this Convention. Thus it is possible for inventors (or the assignees of such inventors) who have applications pending in the United States Patent Office, to file corresponding applications in other countries which are members of the Convention and obtain the benefits of the Convention. If the invention covered by the United States application has been in public use, or has been described in a printed publication, before the date of filing of such application, such provisions of the Convention would be of no benefit in some of the foreign countries where use or publication of the invention prior to the effective filing date of the foreign application invalidates a patent issuing from such application.

While it is possible under certain conditions to file a patent application in a foreign country after the expiration of one year from the date of filing of the corresponding application in the United States, the foreign application in that case becomes a *non-Convention* application and is not entitled to the priority date of the U. S. application. This means that if there has been any publication or public use of the invention prior to the date of filing of the application in the foreign country, the patent in that country may be held to be invalid. In some countries, for example Great Britain and Australia, the publication or use of the invention must take place in that country to invalidate the patent. Other countries, France and Italy, for example, have a stricter law which provides that publication or public use of the invention *anywhere* prior to the effective filing date of the application in such country, will serve to invalidate the patent.

Some Related U. S. Applications Can Be Combined Into One

In most of the foreign countries, it is possible for a corporation to file applications in its own name covering inventions which have been assigned to it by the inventors. One advantage of this is that since the foreign applications do not have to be filed in the name of the inventor, two or more related applications owned by the same company may be combined into a single application providing that there is unity of invention in the subject matter of such applications. This action may reduce considerably the expense of filing and maintenance of the foreign patents. This is unlike the situation in the United States and in Canada. In these countries, if different inventors, working individually rather than jointly, make inventions on the same general subject matter, separate applications must be filed in the names of the individual inventors to cover the various features or improvements made by each.

There are, however, possible disadvantages which may arise in some cases as a result of combining two or more U. S. applications into a single foreign application. In many of the foreign countries, the patent must contain a main claim which points out the characteristic features of the invention, and all of the other claims in the patent defining other aspects of the invention must be dependent upon such main claim. In some cases, therefore, the protection afforded by the foreign patent which combines the disclosures of two or more U. S. applications may be somewhat more limited than would be the case if the inventions had been filed on separately. The question of whether it is advisable to combine, or to file separately in the foreign countries, is therefore one which needs careful consideration.

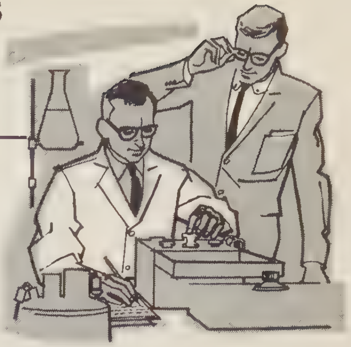
Some of the foreign countries have examination systems similar in some respects to that of the United States Patent Office, that is, they make a search through prior patents and publications to determine whether the applicant is entitled to a patent. In some of these countries, for example Western Germany, Holland and Sweden, the examination is quite strict. On the other hand, France and certain other countries require no examination of the application as to novelty and no claims defining the invention. These countries require simply a summary or resume of the invention, and in an infringement action, it is left to the courts to determine the scope of the protection afforded by the patent.

Taxes Must Be Paid During the Life of Many Foreign Patents

In substantially all foreign countries—Canada is an exception—there are taxes or maintenance fees which must be paid in order to keep the patent in force for its full life, which varies from 15 to 20 years in the different countries. In some countries, the taxes or maintenance fees become payable as soon as the patent is granted. In others, the fees may become payable at the beginning of some later year, such as the second to the fifth year after grant of the patent. The general rule is that the annual taxes or maintenance fees increase as the patent becomes older. One object in having the taxes or maintenance fees increase each year is to encourage lapsing of patents if the owner is not using the invention covered by it or has not licensed others under it. In some countries the rate of increase of taxes or maintenance fees is very rapid particularly toward the end of the life of the patent. For example, in Western Germany the taxes which start at the third year amount to the equivalent of about $12. By the tenth year the tax has increased so that the amount is about $95, by the fifteenth year to $280, and in the eighteenth or final year to about $407. In addition to the above amounts there is a relatively small service charge of the foreign patent attorney who attends to the details of paying the tax. In all of the other foreign countries, however, the taxes or maintenance fees are considerably lower than those of Western Germany.

Patents in most foreign countries differ in at least one other important respect from those in the United States. In general, the laws require *working* of the invention. In many countries the penalty for *non-working* is that after the expiration of a specified number of years, the patent is subject to compulsory licensing or revocation for non-working. In some countries the granting of a license under the patent may be considered as compliance with the working requirements. If the patent actually is not being worked in the country and there have been no licenses granted under it, it is possible in some cases to comply with the working requirements by furnishing proof that products embodying the invention have been recently imported into the country involved. In some of the foreign countries, if proof of actual working or working by importation cannot be furnished, it is possible to effect *nominal working*. This can be done by publishing advertisements in the country involved offering licenses to the public upon mutually satisfactory terms or, if no agreement can be reached between the licensor and the would-be licensee, upon terms decided upon by the Patent Office.

Notes About Inventions and Inventors



The following is a general listing of patents granted in the names of General Motors employes during the period April 1, 1958 to June 30, 1958.

Contributed by

Patent Section

Detroit Office

AC Spark Plug Division
Flint, Michigan

- **Karl Schwartzwalder,** (*B.Cer.E., 1930, and M.S., 1931, The Ohio State University*) director of research, Ceramic Laboratory, and **Arthur V. Somers,** (*B.S. in chemistry, Michigan State University*) senior experimental chemist, Research and Development Laboratory, inventors in patent 2,829,063 for sealing cement for spark plugs.

- **Earl M. Brohl,** senior project engineer, Automotive Engineering Department, and **Robert N. Falge,** formerly with Guide Lamp Division, now retired, inventors in patent 2,831,262 for a projection lamp aiming device.

- **Fortunato Barberis,** (*M.M.E., 1922, and Aero.E., 1923, Royal Polytechnical, Torino, Italy*) project engineer, Spark Plug Engineering Department, inventor in patents 2,831,322 and 2,835,110 for an injector ignitor plug.

- **James A. Norton,** (*Associate in Science degree, Flint Junior College, 1934; S.B. degree, University of Chicago, 1936; Ph.D. degree in organic chemistry, University of Chicago, 1939*) senior experimental chemist, Research Department, inventor in patent 2,834,983 for an apparatus for fabrication of perforated tubes.

- **Karl Schwartzwalder*, and Robert W. Smith,** (*Ph.D., University of Michigan, 1933*) supervisor, Physics and Metallurgy Laboratories, inventors in patent 2,837,679 for a glass sealed centerwire structure.

***Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.**

- **Gordon B. Elder,** (*B.S.Aero.E., University of Michigan, 1937*) staff engineer, Fuel Pump Engineering Department, and **Gordon W. Harry,** (*B.S.M.E., University of Michigan, 1923*) staff engineer, Automotive Products Engineering Department, inventors in patent 2,840,002 for fuel pumps of the diaphragm type.

- **Urban A. Weber, Jr.,** (*B.M.E., Marquette University, 1957*) senior project engineer, Bombing Navigational Systems Engineering Department, inventor in patent 2,840,404 for an adjustable fastening device.

Allison Division
Indianapolis, Indiana

- **Arthur W. Gardiner,** (*A.B. degree, Swarthmore College, 1920*), special assignment, Advanced Design and Development Section, inventor in patent 2,828,602 for a multiflap variable nozzle.

- **Kenneth O. Johnson,** (*B.S.Aero.E., Purdue University, 1950*) senior project engineer, Aero Thermo Design Department, inventor in patent 2,828,607 for a dual engine support.

- **Esten W. Spears, Jr.,** (*B.S.M.E., University of Kentucky, 1942*) senior project engineer, turbo-jets, inventor in patent 2,832,528 for an alcohol mist icing prevention.

- **Victor W. Peterson,** (*B.S.M.E., Rose Polytechnic Institute, 1939*) Advanced Design and Development Section, and **Marshall H. Thomas,** (*M.E., Kansas State University, 1939*) supervisor—reduction gears, Power Turbines Department, inventors in patent 2,833,385 for a clutch control with cooling means therefor.

- **Chester E. Hockert,** (*B.S.M.E., Armour Institute, 1937, and M.S.M.E., Cornell University, 1941*) chief engineer, Turbo-Jet Engineering Department, inventor in patent 2,835,342 for air inlet screens.

- **John R. Hayes,** (*General Motors Institute, 1944*) senior project engineer, Turbine Engine Engineering Department, inventor in patent 2,836,391 for a turbine bucket with cast-in insert.

- **Charles J. McDowall,** (*B.S.M.E., University of Florida, 1927*) technical assistant to the director of engineering, and **Oscar V. Montieth,** (*B.S.E.E., New Mexico College of Agriculture and Mechanic Arts, 1931*) chief development engineer, Production Design and Development Group, inventors in patent 2,836,959 for a gas turbine engine supporting frame.

- **Arthur W. Gaubatz,** (*B.S., University of Wisconsin, 1920*) senior project engineer, Experimental Engineering Department, inventor in patent 2,836,987 for a mechanical discriminating device.

- **Edward L. Bolin,** (*B.S. in chemistry, Butler University, 1933*) plant metallurgist, Metallurgy Department, inventor in patent 2,836,510 for nickel plating by chemical reduction.

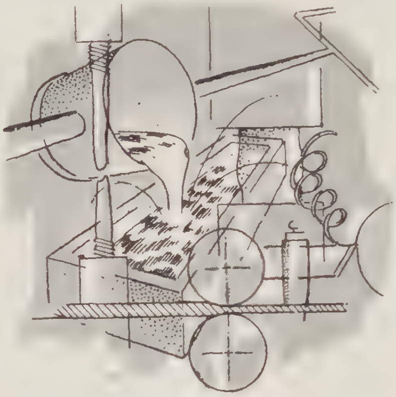
- **Charles F. Hayes,** (*General Motors Institute, 1943*) head of power turbine combustion, Turbine Fuel Systems and Components Engineering Department, and **Leroy W. Shutts,** product engineer, inventors in patent 2,839,894 for a supporting arrangement for a gas turbine combustion chamber.

- **William S. Castle,** (*B.S., University of Illinois, 1942*) group leader on turbines and compressors, Jet Engine Design Engineering Department, and **John M. Wetzler,** (*B.S.M.E., The Johns Hopkins University, 1939*) section head, Turbo-Jet Design Group, inventors in patent 2,840,298 for a heated compressor vane.

- **Gilbert E. Chapman,** (*B.S.M.E., 1943, and M.S.M.E., 1948, Purdue University*) assistant chief—preliminary design air breathing engines, Aircraft Engineering Department, inventor in patent 2,837,270 for an axial flow compressor.

Aeroproducts Operations
Allison Division
Vandalia, Ohio

- **Alan B. Blackburn,** (*Marietta College and The Ohio State University*) experimental engineer, Design Engineering Department, inventor in patent 2,831,992 for a permanent magnet rotor assembly.



- **Kenneth L. Berninger,** (*Purdue University*) senior project engineer, **Darrell E. Royer,** (*Sinclair College and University of Dayton*) senior designer, Engineering Department, and **William A. Weis,** (*B.M.E., University of Dayton, 1938*) senior designer, inventors in patent 2,839,070 for a governor valve assembly.

Buick Motor Division
Flint, Michigan

- **Joseph D. Turlay,** (*B.S.M.E., Oregon State College, 1928*) director, power plant activities, inventor in patents 2,829,017 and 2,838,941 for a piston and an internal combustion engine counterweight and cylinder construction, respectively.

- **Lloyd E. Muller,** (*B.S.M.E., University of Kansas, 1929*) director,. experimental engineering, inventor in patent 2,836,257 for an air intake device.

- **Charles D. Holton,** (*B.S.M.E., University of Michigan, 1931*) engineer, Engineering Department, inventor in patent 2,840,195 for a brake drum.

- **Rudolph J. Gorsky,** (*General Motors Institute, 1935*) staff engineer, Transmission Division, Engineering Department, inventor in patent 2,834,445 for a sprag clutch.

Cadillac Motor Car Division
Detroit, Michigan

- **Casimer J. Cislo,** (*Wayne State University and University of Michigan*) senior project engineer, inventor in patent 2,829,212 for a steering wheel assembly.

- **Daniel M. Adams,** (*Detroit Institute of Technology and University of Michigan*) staff engineer, Body and Sheet Metal Section, inventor in patent 2,831,961 for a motor vehicle fender and headlamp assembly.

- **Bruce Edsall,** (*B.S.M.E., Wayne State University, 1942*) staff engineer in charge of transmissions, rear axles, propeller shafts and brakes, Engineering Department, inventor in patent 2,832,231 for accumulator controls for automatic transmission.

- **Edgar Haigh,** (*B.S.E., Halifax Technical College, Yorks, England, 1923*) staff engineer, Fuel and Exhaust Department, and **Carlton A. Rasmussen,** (*B.S.M.E., Purdue University, 1950*) assistant chief engineer, inventors in patent 2,832,328 for an intake manifold.

- **Jules A. Olivier,** (*Mechanics Institute, New York*) senior project engineer, Engineering Department, and **William J. Tell,** retired, inventors in patent 2,833,593 for a convertible top with an inflatable covering.

- **Charles F. Arnold,** (*B.S.M.E., University of Cincinnati, 1923*) chief engineer, and **Webster J. Owen,** (*B.S., Beloit College, 1936*) assistant staff engineer, inventors in patent 2,837,288 for an automotive heating, ventilating, and defrosting system.

- **Ralph H. Johnson,** (*B.S.M.E., University of Washington, 1952*) senior project engineer—engines, inventor in patent 2,837,384 for a piston.

Central Foundry Division
Saginaw, Michigan

- **Herman A. Laurenz**, experimental engineer, Experimental and Development Department, and **Kenneth E. Spray**, (*B.S.M.E., Michigan State University, and Muskegon Community College*) senior project engineer, Experimental and Development Department, inventors in patent 2,832,112 for a shell mold casting and method.

Chevrolet Motor Division
Detroit, Michigan

- **Adelbert E. Kolbe**, (*University of Michigan*) assistant staff engineer in charge of advanced engine design, Research and Development Department, inventor in patent 2,828,726 for a carburetor mounting device.

- **Robert E. Seelye**, (*Michigan State University*) senior process engineer, Master Mechanics Department, Spring and Bumper Plant—Livonia, Michigan, inventor in patent 2,828,998 for a work jaw means.

- **Adelbert E. Kolbe*, and George J. Philips**, general superintendent, Pattern Shop, Saginaw Grey Iron Foundry Plant, inventors in patent 2,831,225 for a method of making cylinder blocks.

- **E. J. Premo**, (*Chrysler Institute*) executive assistant chief engineer, Engineering Department, inventor in patent 2,832,397 for an interconnected seat cushion and back.

- **Harry R. Urban**, (*Fenn College*) design group leader, Drafting Department, inventor in patent 2,837,285 for a thermostatic and pressure responsive control valve.

- **John P. Hobart**, (*E.E., University of Cincinnati, 1921*) design engineer, Truck Chassis Department, inventor in patent 2,839,319 for a rod clamp.

- **Godfrey Burrows**, master mechanic, Flint Manufacturing Plant, inventor in patent 2,840,388 for frame members.

Delco Radio Division
Kokomo, Indiana

- **William R. Kearney**, (*B.S.M.E., Purdue University, 1933*) senior project engineer, Mechanical Engineering Section, inventor in patent 2,832,889 for a signal seeking radio control.

- **James H. Guyton**, (*B.S.E.E., 1934, and M.S.E.E., 1935, Washington University*) chief engineer—radio, Engineering Department, inventor in patent 2,835,847 for an automatic headlamp dimmer system.

- **Bertram A. Schwarz**, technical assistant to the general manager, and **Manfred G. Wright**, (*B.S.M.E., Purdue University, 1938*) head, Mechanical Engineering Section, inventors in patent 2,835,817 for a motor operated tuning means.

Delco-Remy Division
Anderson, Indiana

- **Charles E. Bates**, (*B.S.Met.E., University of Illinois, 1943*) engineer, Advance Engineering Department, inventor in patent 2,831,910 for a storage battery.

- **John E. Buskirk**, (*B.S.Ch.E., The Ohio State University, 1948*) engineer, Advance Engineering Department, inventor in patent 2,835,720 for a battery vent plug.

- **Don E. Key**, chief machine designer, Process Engineering Department, inventor in patent 2,836,018 for a manufacturing device.

- **William C. Edmundson**, (*B.S.E.E., Purdue University, 1934*) staff engineer, Product Engineering Department, inventor in patent 2,837,934 for an adjustable linkage.

- **Verle E. McCarty**, (*Purdue University*) engineer, Engineering Department, inventor in patent 2,839,186 for a distributor structure package.

- **Lyman A. Rice**, (*B.S.E.E., University of Utah, 1935, and M.S.E., University of Michigan, 1936*) staff engineer, inventor in patent 2,839,746 for an electromagnetic device.

- **Herman L. Hartzell**, (*B.E.E., The Ohio State University, 1924*) chief engineer, Engineering Department; **Perry W. House**, (*B.S.Ch.E., Purdue University, 1933*) assistant chief engineer, Engineering Department; **Clarence E. Tice**, (*B.S.M.E., Purdue University, 1933*) master mechanic; and **Clayton W. Smith**, checker and layout man, Engineering Drafting Department, inventors in patent 2,839,935 for an engine starter.

These patent listings are informative only and are not intended to define the coverage which is determined by the claims of each one.

Detroit Diesel Engine Division
Detroit, Michigan

- **Charles H. Frick**, (*B.S., Iowa State College, 1934*) senior project engineer, Engineering Department, and **Vernon E. Schaefer, Jr.**, (*B.S., University of Michigan, 1938, and M.S., Case Institute of Technology, 1939*) staff engineer, production and field problems, Engineering Department, inventors in patent 2,836,410 for a servo control mechanism for a governor.

- **John Dickson**, (*diploma, Royal Technical College, Glasgow, Scotland*) staff engineer in charge of forward design, Engineering Department, inventor in patent 2,839,041 for a valve for internal combustion engine.

- **Harold G. Haines**, head, generator sets, Product Application Group, inventor in patent 2,840,314 for a heating system.

Detroit Transmission Division
Ypsilanti, Michigan

- **Walter B. Herndon**, (*B.S.E., State College of Washington, 1928, and M.S.E., University of Michigan, 1930*) director of engineering and sales, inventor in patent 2,831,365 for a transmission control system.

Diesel Equipment Division
Grand Rapids, Michigan

- **Ellen L. Karchner**, (*B.S. in chemistry, Mary Manse College, 1941*) experimental chemist, Metallurgical Department, inventor in patent 2,832,708 for a method of treating stainless steel to improve its corrosion resistance.

- **Merton V. Albaugh,** senior process engineer, Manufacturing Process Department, inventor in patent 2,838,931 for a machine for testing hydraulic tappets.

Electro-Motive Division
La Grange, Illinois

- **William F. Holin,** (*M.E., Konstanz, Germany*) senior project engineer, inventor in patents 2,829,015 and 2,833,377 for a bearing assembly and a railway vehicle brake rigging, respectively.

- **Torsten O. Liliquist,** electrical research engineer, inventor in patent 2,831,464 for a fluid pressure motor.

- **Thomas B. Dilworth,** assistant chief engineer, Engineering Department, inventor in patent 2,840,009 for a railway vehicle truck.

- **Edward P. Kupka,** (*B.A. in mathematics and physics, Coe College, 1935, and B.S.E.E., Iowa State College, 1937*) senior project engineer, Research Department, inventor in patent 2,840,731 for a variable angle magnetic drive.

GM Engineering Staff
Detroit, Michigan

- **Lewis D. Burch,** (*B.S.M.E., Purdue University, 1923; George Washington University; Akron Law School; and Wayne State University*) patent attorney, Patent Section, Detroit Office, and **Herman L. Hartzell,** (*B.E.E., The Ohio State University, 1924*) chief engineer, Engineering Department, Delco-Remy Division, inventors in patent 2,828,725 for an engine governor.

- **Von D. Polhemus,** (*B.S.M.E., University of Cincinnati, 1933*) engineer in charge, Structure and Suspension Development Group, inventor in patent 2,833,552 for a dynamic vibration damper.

- **Donald D. Miller,** (*Ferris Institute and University of Michigan*) section engineer, Passenger Car Development Group, and **Gilbert H. Wolf,** (*International Correspondence School*) shop foreman, Passenger Car Development Group, inventors in patent 2,835,183 for a vehicle air circulating system.

- **William K. Steinhagen,** (*B.S.E., 1947; M.S.E., 1948, University of Michigan*) assistant engineer in charge, Power Development Group; **Philip L. Francis,** project engineer, Power Development Group; and **John Dolza,** no longer with GM, inventors in patent 2,835,436 for a refrigerating apparatus.

- **Albert J. Zupancic,** senior technician, Test Facilities Department; inventor in patent 2,837,072 for a fuel metering system.

- **George P. Ransom,** (*B.S.M.E., University of Michigan, 1949*) section engineer, Power Development Group, inventor in patent 2,837,074 for a fuel injection system.

- **Ralph S. Johnson,** (*B.S.M.E., Purdue University, 1933*) senior project engineer, Power Development Group, inventor in patent 2,838,957 for an engine balancing means.

- **Charles A. Chayne,** (*B.S.M.E., Massachusetts Institute of Technology, and Harvard University, 1919*) vice president in charge of GM Engineering Staff, and **Maurice A. Thorne,** (*B.A., George Washington University, and University of Maryland*) engineer in charge, Vehicle Development Group, inventors in patent 2,840,368 for a torsion spring rear suspension.

Euclid Division
Cleveland, Ohio

- **Walter F. Double,** development engineer, Engineering Department, inventor in patent 2,835,137 for a transmission friction clutch arrangement.

- **Russell C. Williams,** (*B.S.M.E., University of Illinois, 1932*) manager, research and test, Engineering Department; **Edwin J. Selyem,** (*Fenn College, and Case Institute of Technology*) now senior designer, Cleveland Ordnance Plant, Cadillac Motor Car Division; and **John P. Carroll,** (*Bradley University*) chief product engineer, Engineering Department, inventors in patent 2,837,378 for a hydraulic recoil mechanism.

- **Edwin J. Selyem*, and Janis Mazzarins,** (*Technical University, Aachen, West Germany*) senior project engineer, Engineering Department, inventors in patent 2,837,379 for a hydraulic track adjuster.

- **John A. Walko,** (*B.S.M.E., The Ohio State University, 1951*) product engineer, Engineering Department, inventor in patent 2,839,313 for a double articulated rock ejector for vehicle dual wheels.

Fisher Body Division
Detroit, Michigan

- **Thomas J. Hosea,** (*B.M.E., General Motors Institute, 1953*) senior designer, Design and Drafting Department, inventor in patent 2,822,568 for a deck lid hinge with adjustable spring counterbalance.

- **Charles J. Griswold, Jr.,** (*B.S. in engineering mechanics, Purdue University, 1953*) senior project engineer, Product Engineering, Design and Drafting Department, and **Harold E. VanVoorhees,** (*B.S.M.E., Purdue University, 1915*) senior design engineer, New Product Development Department, Ternstedt Division, inventors in patent 2,822,203 for a door latch and control means.

- **Charles W. Rantala,** (*University of Detroit*) designer, Design and Drafting Department, inventor in patent 2,821,429 for a vehicle body drainage and sealing means.

- **William H. Webber,** metal worker engineering leader, inventor in patent 2,822,031 for a seat spring construction.

- **Stanley Duluk,** (*B.S.M.E., Wayne State University, 1949*) senior designer, Experimental and Development Department; **Louis P. Garvey,** (*B.M.E., University of Detroit, 1939*) senior project engineer, Experimental and Development Department; and **Clyde H. Schamal,** (*B.S.E.E., University of Notre Dame, 1927*) engineer in charge, Central Experimental and Development Department, inventors in patent 2,820,506 for a vehicle seat.

- **Russell N. Blanton,** body design group leader, Design and Drafting Department, and **Guy L. Tucker,** (*Tri-State College*) assistant engineer in charge, door department, Body Engineering Activity, inventors in patent 2,822,215 for a door window roller guide assembly.

- **Clarence P. McClelland,** (*University of Detroit*) senior project engineer, Design and Drafting Department, inventor in patent 2,824,456 for a flexible cable drive for oscillating shafts.

- **Thomas W. Shearer, Jr.,** (*B.S.E.E., Lawrence Institute of Technology, 1943*) senior production engineer, Production Engineering Activity, Process Development Department, and **Arthur F. Hessler,** senior production engineer, Process Development Department, inventors in patent 2,824,213 for a consumable electrode welding method and mechanism.

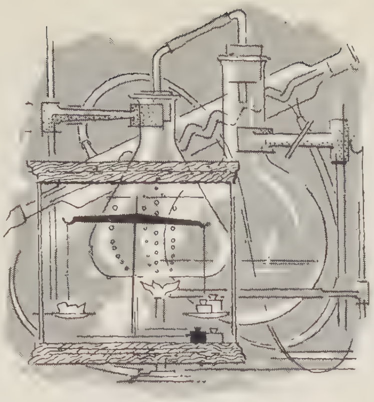
- **James D. Leslie,** (*B.M.E., University of Detroit, 1939*) engineer in charge, Mechanical Department, **Clyde H. Schamel*,** and **Harold E. Van Voorhees*,** inventors in patent 2,827,321 for a door latch for a pillarless automobile.

- **Robert M. Fox,** (*B.M.E., General Motors Institute, 1950*) senior project engineer, Experimental and Development Department, and **Russell A. Grout,** senior layout man, Experimental and Development Department, inventors in patent 2,828,157 for a water shield for automobile door latch.

- **Joseph E. Kubaka,** (*Columbia University, Detroit Institute of Technology and Wayne State University*) senior engineer in charge, Experimental and Development Department, and **Alfons A. Limberg,** (*Automotive Engineering School, Berlin, Germany*) now chief engineer on Body and Truck Exterior, GM Styling Staff, inventors in patent 2,817,558 for a roof rail weatherstrip for convertible type doors.

- **Napoleon P. Boretti,** (*B.E.E., University of Detroit, 1935*) assistant engineer in charge, Process Development Department, and **Ernest V. Harper,** (*B.S.E., University of Marquette and Northwestern University, 1946*) production engineer, Mechanical Handling—Production Engineering Activity, inventors in patent 2,838,831 for a fluid cleaner.

- **Clyde H. Schamel*, and Claud S. Semar,** (*Detroit City College, University of Michigan, and Wayne State University*) senior project engineer, Experimental and Development Department, inventors in patent 2,828,999 for a window arrangement for vehicle bodies.



- **Napoleon P. Boretti*,** inventor in patent 2,829,239 for a welding electrode comprising a shank and removable cap.

- **Andrew E. Hallek,** engineer in charge, convertible styles, Production Engineering Activity, and **Alfred E. Wittke,** project engineer, Production Engineering Activity, inventors in patent 2,831,718 for a convertible top header lock assembly.

- **Joseph G. Joachim,** senior project engineer, Experimental and Development—Production Engineering Activity, and **Alfred B. Sauer,** (*B.S.E., University of Michigan, 1952*) senior layout draftsman, Experimental and Development—Production Engineering Activity, inventors in patent 2,833,536 for a power operated rear compartment actuator and lock assembly.

- **Englebert A. Meyer,** senior project engineer, Engineering Department, inventor in patent 2,836,458 for a retaining means for door trim panel.

- **Joseph B. Griffo,** (*B.M.E., General Motors Institute, 1957*) senior detail draftsman, Experimental and Development Department, now on military leave, inventor in patent 2,837,143 for a border wire interlock for sinuous spring cushion.

Frigidaire Division
Dayton, Ohio

- **Verlos G. Sharpe,** (*B.S.M.E., Purdue University, 1948*) section engineer, Refrigerated Appliances Engineering Department, inventor in patent 2,829,506 for an ice block harvesting device; in patent 2,840,438 for an adjustable refrigerator shelf; and in patent 2,840,439 for a refrigerator shelf.

- **Millard E. Fry,** (*B.S.M.E., University of Pittsburgh, 1931*) senior project engineer, Non-Refrigerated Appliances Engineering Department, inventor in patents 2,831,956 and 2,836,698 for a domestic appliance.

- **Verlos G. Sharpe*, and Everett C. Armentrout,** (*B.S. in physics, University of Dayton, 1957*) project engineer, Refrigerated Appliances Engineering Department, inventors in patent 2,835,113 for a combined ice block releasing and storing bucket.

- **Carl M. Schell,** (*Wittenberg College, University of Chicago*) general supervisor of drafting, Research and Future Products Department, inventor in patent 2,835,187 for an air grill for a refrigerating apparatus.

- **Robert D. Bremer,** (*B.S.E.E., Purdue University, 1934*) senior project engineer, Non-Refrigerated Appliances Engineering Department, inventor in patent 2,835,780 for a domestic appliance.

- **Harry F. Clark,** (*E.E., University of Cincinnati, 1921*) senior project engineer, Refrigerated Appliances Engineering Department, inventor in patent 2,836,781 for an electrical apparatus.

- **Jesse L. Evans,** (*B.S.M.E., University of Dayton, 1943*) senior project engineer, Engineering Department, inventor in patent 2,836,268 for a door construction.

- **Orson V. Saunders,** supervisor of major product line, Refrigerated Appliances Engineering Department, inventor in patent 2,837,816 for a method of manufacturing single and double doors.

- **Lester M. Miller,** (*Dayton Art Institute, Art Academy of Cincinnati, and the Central Academy of Commercial Art*) junior engineer, Engineering Department, inventor in patent 2,838,357 for a vertically adjustable refrigerator shelf.

GMC Truck and Coach Division
Pontiac, Michigan

- **Ferdinand R. Eichner,** superintendent, Experimental Sheet Metal Department, inventor in patent 2,837,363 for a latching mechanism.

Guide Lamp Division
Anderson, Indiana

- **Charles W. Miller,** (*Purdue University*) project engineer, Engineering Department, and **Harold E. Todd,** (*B.S.E.E., Purdue University, 1940*) senior project engineer—factory and field service problems, Engineering Department, inventors in patent 2,829,307 for an oscillator controlled automatic headlamp dimmer system.

- **John H. Diedring,** (*Fenn College*) engineering head, original equipment lamps, inventor in patent 2,831,964 for a water drainage from vehicle lamps.

- **Eugene G. Matkins,** (*B.M.E., General Motors Institute, 1952*) project engineer, Engineering Laboratory, and **Charles W. Miller***, inventors in patent 2,832,914 for a condition responsive control circuit.

- **David P. Clayton,** designer, Product Engineering Department, and **Robert N. Falge,** retired, inventors in patent 2,837,609 for an electric switch.

Harrison Radiator Division
Lockport, New York

- **John R. Holmes,** retired; **William J. DeBeaubien,** (*General Motors Institute, 1932*) heating, ventilating and air conditioning engineer, Pontiac Motor Division; and **Herman S. Kaiser,** body engineer, Engineering Department, Pontiac Motor Division, inventors in patent 2,831,327 for vehicle air conditioning systems.

- **Charles S. Bailey,** (*B.M.E., Clarkson College, 1936*) project engineer, inventor in patent 2,837,284 for an automobile radiator thermostat.

Inland Manufacturing Division
Dayton, Ohio

- **Cletus L. Moorman,** (*University of Dayton*) project engineer, and **Ellwood F. Riesing,** deceased, inventors in patents 2,830,832 and 2,830,858 for a fluid seal.

- **John H. Gates,** (*B.I.E., General Motors Institute, 1948*) supervisor of plant layout, Process Engineering Department, and **Vernon B. Burden,** (*General Motors Institute, 1942*) process engineer, Process Engineering Department, inventors in patent 2,835,777 for a heated steering wheel.

- **Paul E. Clingman,** (*General Motors Institute, 1935*) supervisor, Quality and Control Distribution, inventor in patent 2,838,436 for a method of bonding.

Moraine Products Division
Dayton, Ohio

- **William F. Erickson,** (*General Motors Institute, 1943*) section engineer, Engineering Department, and **DeLoss D. Wallace,** (*B.M.E., The Ohio State University, 1926*) section engineer, Engineering Department, inventors in patent 2,830,542 for a fluid pump.

- **Frederick W. Sampson,** (*M.E., Cornell University, 1924*) section engineer on special assignment, and **Allen L. Everitt,** (*M.S.M.E., Purdue University, 1931*) now section engineer, Engineering Department, Inland Manufacturing Division, inventors in patent 2,831,404 for a recoil buffer for guns.

- **James O. Helvern,** (*B.A., Wittenburg College, 1927*) senior designer, inventor in patent 2,832,444 for a brake booster.

- **Jean A. Lignian,** (*B.S.E.E., University of Michigan, 1928*) chief inspector, inventor in patent 2,840,196 for a brake shoe.

New Departure Division
Bristol, Connecticut

- **Leland D. Cobb,** (*E.E., Pratt Institute, 1928*) manager, Research and Development Laboratory, inventor in patent 2,837,188 for a one way clutch.

- **Rene E. Sauzedde,** (*University of Michigan, 1929, and New York University*) manager, bicycle and sprag products, Product Engineering Department, inventor in patent 2,837,189 for a sprag clutch.

Oldsmobile Division
Lansing, Michigan

- **Jack B. Ridenour,** (*B.S.M.E., Michigan State University, 1948*) senior project engineer, body section, Product Engineering Department, inventor in patent 2,838,985 for a pivotal mounting device.

- **Ernest A. Leavengood,** (*B.S. in physics, Michigan State University*) head, Electrical Section, Engineering Department, and **Donald W. Severance,** (*Michigan State University and Massachusetts Institute of Technology*) project engineer, Electrical Section, Engineering Department, inventors in patent 2,829,002 for a seat positioning mechanism.

- **Robert Seyfarth,** (*B.S.M.E., Princeton University, 1936*) senior engineer, engine group, Product Engineering Department, inventor in patent 2,836,113 for a heating and ventilating system or apparatus for vehicles.

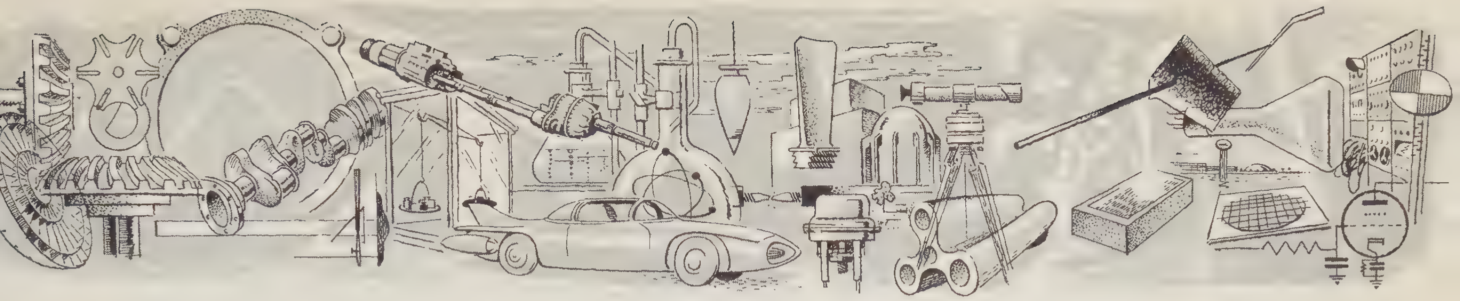
GM Overseas Operations Division
New York, New York

- **Kenneth E. Buckman,** assistant chief engineer, Engineering Department, AC-Delco, General Motors Limited, London, England, inventor in patents 2,835,393 and 2,836,302 for filter elements and fluid filters, respectively.

- **Clifford L. Price,** assistant chief engineer, Engineering Department, AC-Delco, General Motors Limited, London, England, inventor in patent 2,836,258 for a cap for liquid receptacles.

Pontiac Motor Division
Pontiac, Michigan

- **Herman S. Kaiser,** body engineer, Engineering Department, inventor in patent 2,838,989 for a hood latch.



- **Clayton B. Leach,** (*A.B. in mathematics and chemistry, Park College, 1934, and General Motors Institute*) chassis engineer, Engineering Department, inventor in patent 2,837,075 for an engine crankcase structure.

GM Research Laboratories
Detroit, Michigan

- **Joseph F. Lash,** (*B.S.M.E., Michigan State University, 1938*) supervisor, Special Problems Department, inventor in patent 2,828,911 for an unbalance data correction apparatus.

- **Archie D. McDuffie,** (*General Motors Institute, 1934*) now with Buick Motor Division, inventor in patent 2,828,730 for an engine.

- **Walter E. Sargeant,** (*B.S.E.E., University of Michigan, 1926*) senior research engineer, Physics Department, inventor in patent 2,829,324 for a low voltage electrical motive means.

- **Robert F. Thomson,** (*B.S.M.E., 1937; M.S.M.E., 1940; and Ph.D., 1941, University of Michigan*) head, Metallurgical Engineering Department, inventor in patent 2,831,243 for a sintered powdered copper base bearing.

- **Alfred W. Schlucter,** (*B.S.Ch.E., 1919; M.S.Ch.E., 1921; Ph.D., 1926, University of Michigan*) research engineer, Mechanical Development Department, inventor in patent 2,831,764 for a bearing.

- **Warren H. Smith,** (*B.S.M.E., Purdue University, 1928*) supervisor of design, Mechanical Development Department, inventor in patent 2,839,911 for a free piston engine.

Saginaw Steering Gear Division
Saginaw, Michigan

- **Robert P. Rohde,** (*B.S.M.E., University of Michigan, 1950*) senior project engineer, and **Philip B. Zeigler,** (*B.S.E., Purdue University, 1941*) chief engineer, inventors in patent 2,828,616 for a power transmitting element with damping means.

- **Arthur F. Bohnhoff,** (*B.S.M.E., Michigan College of Mining and Technology, 1938*) senior designer, and **Charles W. Spalding,** (*B.S.M.E., Michigan State University, 1940*) designer, inventors in patent 2,828,722 for a windshield wiper motor.

- **W. Blair Thompson,** (*B.M.E., General Motors Institute, 1950*) assistant chief engineer; **Philip B. Zeigler***; and **C. W. Lincoln,** retired, inventors in patent 2,828,829 for fluid power steering.

- **David A. Galonska,** staff engineer, inventor in patent 2,836,075 for a one way ball nut.

- **Earl W. Glover,** designer, inventor in patent 2,839,902 for an indexing means for splined connected members and the like.

GM Styling Staff
Detroit, Michigan

- **Donald D. Hoagg,** (*Pratt Institute, 1949*) chief designer, Design Development Studio, and **John W. Yee,** (*M.E., Wayne State University*) general supervisor, Exterior Ornamentation Group, Exterior Engineering Department, inventors in patent 2,829,241 for an automotive vehicle lamp.

Ternstedt Division
Detroit, Michigan

- **Michael A. DeAngelo,** gauge designer, Inspection Department, inventor in patent 2,828,996 for a striker for a door latch.

- **Bewley D. Priestman,** (*B.A. in engineering science, 1938, and M.A.M.E., 1937, University of Cambridge, England*) engineering group supervisor, Product Engineering and Development Department, inventor in patent 2,832,626 for a door latch—adjustable linkage for push button.

- **Emmett A. Bartlow,** superintendent, Maintenance Department, inventor in patent 2,835,158 for a fusible washer with means to protect threads from molten metal.

- **Harold E. Van Voorhees,** (*B.S.M.E., Purdue University, 1915*) senior design engineer, New Product Development Department, inventor in patents 2,836,050 and 2,835,526 for an automobile door latch and a door latch, respectively.

- **Barthold F. Meyer,** (*B.S.M.E., Pratt Institute, 1939, and Johns Hopkins University*) supervisor of mechanical design group, Product Engineering Department, inventor in patent 2,837,362 for a door ventilator latch.

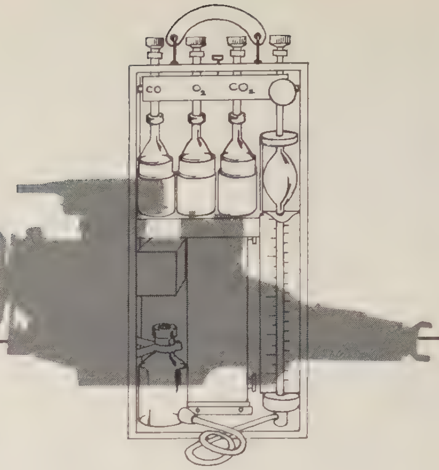
- **Russell G. Corbin,** (*B.S.M.E., University of Michigan, 1950*) design group leader, Product Engineering and Development Department, and **Bewley D. Priestman*,** inventors in patent 2,839,912 for a tail gate latch.

- **James R. Hemeon,** (*M.E., Massachusetts Institute of Technology, 1922*) senior mechanical engineer, Plant Engineering Department, inventor in patent 2,839,895 for a hydraulic driving apparatus.

- **Frank A. Croskey,** research engineer, Process Development Department, and **Charles D. Tuttle,** (*Ph.D., Michigan State University, 1933*) senior experimental chemist, Process Development Department, inventors in patent 2,840,658 for a high tension circuit breaker system.

A Typical General Motors Institute Laboratory Problem:

Develop an Alignment Chart to Find Air-Fuel Ratios from an Orsat Analysis for the Combustion of C_8H_{18}



A fast way to determine the air-fuel ratio of a spark ignition engine is to measure, by means of an Orsat apparatus, two or more constituents of the exhaust gas and then refer to a graphical presentation of exhaust gas composition versus air-fuel ratio. In general, the graphical information does not consider the possibility of both oxygen and carbon monoxide existing simultaneously in the exhaust gas, which is usually the case due to cylinder-to-cylinder variation in fuel distribution and imperfect mixing in the cylinders. The problem presented here is to develop an alignment chart (the simplest type of nomograph) which will allow immediate determination of the air-fuel ratio from an Orsat gas analysis for the combustion of C_8H_{18}. The chart is to be developed based on the assumption that both oxygen and carbon monoxide are present in the exhaust gas.

To EVALUATE the performance and efficiency of a spark ignition engine requires complete information on the air-fuel ratio. In evaluating the air-fuel ratio for one cylinder or the average for a multi-cylinder engine, three methods may be used: (a) establishment of the quantities of air and fuel inducted into the cylinders, (b) analysis of the exhaust products, and (c) utilization of direct reading exhaust gas analyzers.

Method (a) has a disadvantage in that the flow of air to the engine can be affected by the measuring equipment. Also, this method is difficult to carry out when the engine is installed in a moving vehicle. Method (b) utilizes the familiar Orsat apparatus to determine the percentage by volume of the exhaust gas constituents. The quickest method for evaluating air-fuel ratio is method (c), which measures electrically the thermal conductivity of the exhaust gas in comparison with a standard gas of known value. Methods (b) and (c) are simple to perform, involve a minimum of equipment, and are especially valuable during road tests.

The range of air-fuel mixtures which will support combustion extends to both the rich and lean side of the correct air-fuel ratio. This range is usually between 9 lb of air per lb of fuel to 18 lb of air per lb of fuel. The correct air-fuel ratio for complete combustion in a gasoline engine is approximately 15 to 1. Experiments show that maximum power is obtained at an air-fuel ratio of approximately 12.5 to 1. For maximum economy, an air-fuel ratio of approximately 16.5 to 1 is needed.

The range of power requirements for a gasoline engine indicates the magnitude of the problem encountered in arriving at the proper air-fuel ratio. Definite air-fuel ratios are necessary for different conditions of speed and load. It is quite important, therefore, that for testing purposes the air-fuel ratio be determined quickly and accurately.

The problem presented here is based on a typical laboratory project assigned to students enrolled in the applied thermodynamics course at General Motors Institute. Laboratory tests conducted to determine the air-fuel ratio of a spark ignition engine by means of an Orsat analysis indicated the simultaneous presence of CO and O_2 in the exhaust gas. This resulted from cylinder-to-cylinder variation in fuel distribution and imperfect mixing of the fuel within the cylinders. The test results led to the problem of whether an alignment chart could be developed which would take into consideration the fact that both CO and O_2 are present in the exhaust products of the engine. The problem allowed students to apply theoretical analysis and also practical testing procedures by obtaining an actual Orsat gas analysis.

There are several methods available for calculating the air-fuel ratio from an exhaust gas analysis. The most common and convenient method is the carbon-nitrogen balance[1], which can be used for hydrocarbon fuels of known chemical composition. This method assumes that all of the carbon burns to either CO or CO_2 and that no free carbon is present in the exhaust gas. The nitrogen is assumed to be that gas remaining after the various gas constituents have been accounted for by the exhaust analysis.

Problem

The problem is to develop an alignment chart[2] from which can be read the air-fuel ratio when the volume percentage of the various constituents of an exhaust gas analysis is known. The liquid fuel to be used for the exhaust gas analysis has the molecular formula C_8H_{18}. The alignment chart is to be based on the combustion equation of C_8H_{18}. The solution must consider the possibility of both oxygen and carbon monoxide being present in the exhaust gas at the same time.

Nitrogen is to be assumed inert and not to enter into the combustion reaction. The amount of nitrogen in the exhaust, therefore, will be equal to the amount that entered with the air. Air is to be assumed a mechanical mixture composed entirely of nitrogen and oxygen (79 per cent N_2 and 21 per cent O_2 by volume).

Faculty Members-in-Charge:
DONALD R. JENKINS
and FRANK N. CRALL

G.M.I. Cooperative Student:
CHARLES COPELAND
AC Spark Plug Division

Solve a problem dealing with combustion equations and with mathematics of nomographs

The solution to the problem and presentation of the alignment chart will appear in the January-February-March 1959 issue of the GENERAL MOTORS ENGINEERING JOURNAL.

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.... "General Motors Automotive Engine Test Code," Fourth Edition, 1953, pp. 149-150. (The Automotive Engine Test Code is distributed by the GM Research Laboratories, Administrative Engineering Department, Detroit, Michigan.)

D'ALLEVA, B. A., and LOVELL, W. G., "Relation of Exhaust Gas Composition to Air-Fuel Ratio," *S.A.E. Transactions*, Vol. 38, No. 3 (March 1936) pp. 90-6.

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Solution to the Previous General Motors Institute Classroom Problem:

Determine the Static and Total Temperatures of a High Temperature, High Velocity Gas Stream

Faculty Member-in-Charge:
KENNETH W. WOODFIELD

G.M.I. Cooperative Student:
RODGER E. BLOOMFIELD
AC Spark Plug Division

A typical application of heat transfer and fluid dynamics fundamentals

Temperature determination by means of a thermocouple immersed in a high velocity gas stream is somewhat difficult due to friction and stagnation of the gas stream in the immediate vicinity of the thermocouple junction. Since the thermocouple junction experiences radiation and conduction losses, as well as aerodynamic heating, the temperature indicated by the thermocouple is not a true indication of either the static or total temperature of the moving gas stream. It becomes necessary, therefore, to apply basic heat transfer and fluid dynamics fundamentals to arrive at the correct temperatures. This is the solution to the problem presented in the July-August-September 1958 GENERAL MOTORS ENGINEERING JOURNAL. The adiabatic wall temperature is *1,875°F*. The static temperature of the gas stream is *1,842°F* and the total temperature of the gas stream is *1,888°F*.

THE fundamental fact which must be kept in mind in the solution of the problem is the fact that the temperature indicated by the thermocouple immersed in the gas stream is an equilibrium temperature. The temperature so indicated represents a balance between heat losses from the thermocouple junction by the modes of radiation and conduction and heat gain by the mode of convection (Fig. 1). This may be stated verbally as follows:

> heat loss from thermocouple junction by radiation + heat loss from thermocouple junction by conduction along thermocouple wires = heat gain to thermocouple junction by convection.

The above expression may be written symbolically as follows:

$$0.173\, F_\varepsilon F_A A_s \left[\left(\frac{T_w}{100}\right)^4 - \left(\frac{T_e}{100}\right)^4\right] + k_m A_x \left[\frac{(dt)}{(dx)w}\right] = h_e A_s (t_{aw} - t_w) \quad (1)$$

where

F_ε = effective emissivity (dimensionless)
F_A = geometrical configuration factor (dimensionless)
A_s = surface area of thermocouple junction (sq ft)
T_w = absolute temperature of thermocouple junction (°R)
T_e = mean radiant temperature of surroundings (°R)
k_m = mean thermal conductivity o thermocouple wires (Btu per hr-sq ft-°F per ft)
A_x = combined cross sectional area of thermocouple wires (sq ft)
$\frac{(dt)}{(dx)w}$ = temperature gradient along thermocouple wires evaluated at thermocouple junction (°F per ft)
h_e = convective heat transfer coefficient in the presence of aerodynamic heating (Btu per hr-sq ft-°F)
t_{aw} = adiabatic wall temperature (°F)
t_w = temperature of thermocouple junction (°F).

The conduction component

$$q_{cd} = k_m A_x \left[\frac{(dt)}{(dx)w} \right]$$

of equation (1) is difficult to evaluate in practice since the temperature distribution along the thermocouple wires, from the junction to the support, is not known.

The general shape of the temperature distribution curve is known to be concave downward (Fig. 2). The reasons for the concave downward shape of the temperature distribution curve are as follows:

a) The support temperature is lower than either the junction or gas temperature. It follows, therefore, that the difference in temperature (Δt) between the gas and the thermocouple wires must necessarily increase from the junction to the support

b) Heat transfer by convection from the gas to the thermocouple wires is directly proportional to (Δt). This results in an increase in conduction rate along the wires from the junction to the support

c) The absolute value of the localized conduction temperature gradient at the support is greater than the absolute value of the localized conduction temperature gradient at the thermocouple junction.

Because of reasons (a), (b), and (c), the temperature distribution curve must necessarily be concave downward to give a steeper gradient at the support rather than at the junction.

An estimate of the heat loss by conduction from the junction can be made by assuming the temperature distribution along the wires to be linear in nature. Such an estimate would result in a conductive heat loss value considerably higher than would actually exist. The temperature gradient

$$\left(\frac{\Delta t}{\Delta x} \right)$$

for a linear temperature distribution would be steeper than the localized temperature gradient at the thermocouple junction

$$\frac{(dt)}{(dx)w.}$$

That this is evident can be seen by comparing the solid and broken temperature distribution lines shown in Fig. 2.

Working with the temperature gradient in the conduction term gives the following equation:

$$q_{cd} = k_m A_x \left(\frac{\Delta t}{\Delta x} \right)$$

$$q_{cd} = k_m A_x \left(\frac{t_w - t_o}{L} \right) \tag{2}$$

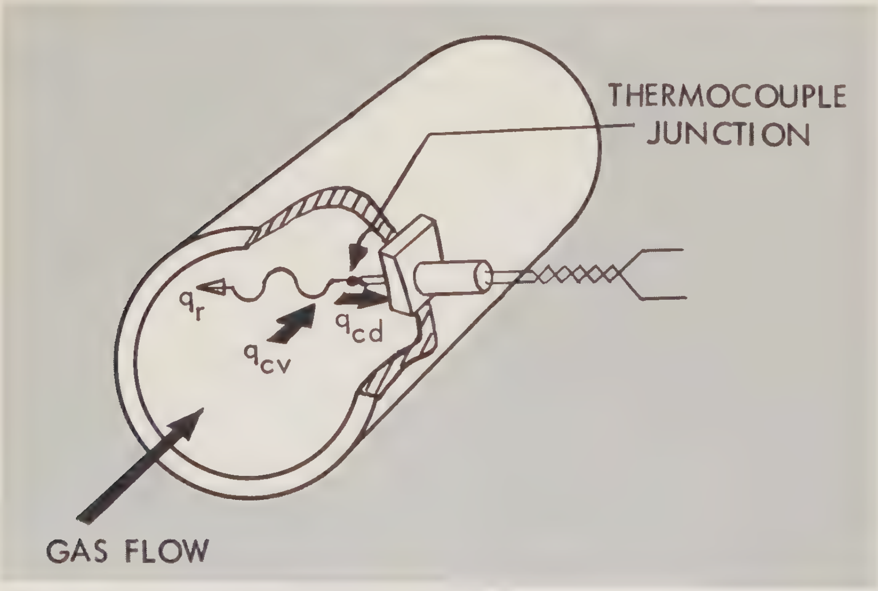


Fig. 1—Shown here is a pictorial representation of heat transfer to and from a thermocouple junction immersed in a high temperature, high velocity gas stream. Heat losses from the junction are by the modes of radiation and conduction. The radiation transfer q_r is a result of net radiant exchange from the junction to the lower temperature surroundings. The conduction transfer q_{cd} occurs along the thermocouple wires from the junction to the lower temperature support. Heat gain to the thermocouple is by the mode of convection q_{cv} from the partially stagnated gas stream to the junction.

where

q_{cd} = rate of heat transfer by conduction (Btu per hr)

$k_m = \frac{k_1 + k_2}{2}$

k_1 = thermal conductivity of chromel wire

k_2 = thermal conductivity of alumel wire

$k_m = \frac{(10 + 28)}{2}$ = 19 Btu per hr-sq ft-°F per ft

$A_x = 2\pi r^2$

r = radius of thermocouple wires

$A_x = 2\pi \left(\frac{0.008}{12} \right)^2 = 2.79(10^{-6})$ sq ft

t_w = 1,700°F

t_o = temperature of thermocouple support = 700°F

L = distance from thermocouple junction to thermocouple support = $\frac{2}{12}$ ft.

Substituting known values into equation (2) gives:

q_{cd} = 0.318 Btu per hr.

The radiation component

$$q_r = 0.173\, F_\varepsilon F_A A_s \left[\left(\frac{T_w}{100} \right)^4 - \left(\frac{T_e}{100} \right)^4 \right] \tag{3}$$

of equation (1) is basically the Stefan-Boltzmann Law of Total Radiation. In the case of heat transfer by radiation between a body and its enclosure, where the area of the enclosure is much greater than the area of the body, the geometrical configuration factor becomes equal to unity and the effective emissivity is equal to the normal total emissivity of the enclosed body[1].

The following known values may be substituted into equation (3) to obtain a value for the radiation component:

F_ε = 0.870

F_A = 1.00

$A_s = 4\pi R^2$

R = radius of sphere formed by thermocouple junction

$A_s = 4\pi \left(\frac{0.035}{12} \right)^2 = 10.7(10^{-5})$ sq ft

T_w = 2,160°R

T_e = 760°R.

Substituting into equation (3) gives

$$q_r = 3.46 \text{ Btu per hr.}$$

A comparison of the radiation component q_r of 3.46 Btu per hr with the conduction component q_{cd} of 0.318 Btu per hr shows the relative insignificance of heat loss by conduction. The conduction loss is less than nine per cent of the total loss. Bearing in mind that the conduction loss as calculated gives a conservatively high value, the actual conduction loss becomes even more insignificant. In fact, the conduction of heat along the thermocouple wire, away from the junction, is reduced if not entirely eliminated by arranging for the gas stream to flow over the bare wires for a given distance after the junction[2].

For the convection component

$$q_{cv} = h_e A_s (t_{aw} - t_w) \qquad (4)$$

of equation (1), the convective heat transfer coefficient h_e must be evaluated for the existing set of operating conditions. For the purpose of correlating heat transfer data in dimensionless form, the heat transfer coefficient h_e is usually written in terms of the Nusselt modulus

$$N_{Nu} = \frac{h_e D}{k_o}$$

where

D = diameter of thermocouple junction (sq ft)

k_o = thermal conductivity of gas stream.

Using this definition of the Nusselt modulus, it can be shown[3] that for steady state conditions and a given geometrical arrangement the Nusselt modulus can be expressed as follows in terms of other dimensionless parameters; namely, the Reynolds Number N_{Re} and the Prandtl Number N_{Pr}:

$$N_{Nu} = f\,(N_{Re}, N_{Pr}).$$

For common thermocouple materials mounted normal to a stream of high velocity air having a range of Mach number from 0.1 to 0.9, a Reynolds Number of from 250 to 30,000, and the diameter of the thermocouple junction as the characteristic diameter, the following dimensionless correlation applies[3]:

$$\frac{h_e D}{k_o} = 0.478 \left(\frac{DV\rho}{\mu}\right)^{0.5} \left(\frac{c_p \mu}{k_o}\right)^{0.3} \qquad (5)$$

where

V = velocity of gas stream (ft per sec)

ρ = density of gas stream (lb per cu ft)

μ = dynamic viscosity of gas stream (lb per hr-ft)

c_p = specific heat at constant pressure of gas stream (Btu per lb-°F).

The heat transfer coefficient h_e for the stated set of conditions, therefore, can be calculated as follows:

$$\left.\begin{array}{lll} D & = & \frac{0.070}{12} \text{ ft} \\ V & = & 800 \text{ fps} \\ \rho & = & 0.0185 \text{ lb per cu ft} \\ \mu & = & 0.1138 \text{ lb per hr-ft} \\ k_o & = & 0.049 \text{ Btu per hr-sq ft-°F per ft} \\ c_p & = & 0.280 \text{ Btu per lb-°F} \end{array}\right\} \text{at } 1{,}700°\text{F}$$

$$N_{Pr} = \frac{c_p \mu}{k_o} = 0.65$$

$$N_{Re} = \frac{DV\rho}{\mu} = 2{,}740.$$

From equation (5):

$$\frac{h_e D}{k_o} = 0.478\,(2{,}740)^{0.5}\,(0.65)^{0.3} = 22.0.$$

Therefore,

$$h_e = 185 \text{ Btu per hr-sq ft-°F.}$$

The adiabatic wall temperature t_{aw} can now be determined as follows by equating equation (4) to the radiation transfer rate obtained from equation (3) and neglecting the conduction transfer rate:

$$h_e A_s (t_{aw} - t_w) = 3.46 \text{ Btu per hr}$$

$$t_{aw} = t_w + \frac{3.46}{h_e A_s}$$

$$t_{aw} = 1{,}700 + \frac{3.46}{185(10.7)\ (10^{-5})}$$

$$t_{aw} = 1{,}875°\text{F}.$$

From the definitions of recovery factor N_{rf} and temperature rise for isentropic stagnation given in the statement to the problem, the static temperature of the gas stream can be calculated as follows:

$$N_{rf} = \frac{(t_{aw} - t_s)}{(t_t - t_s)} \qquad (6)$$

$$(t_t - t_s) = \frac{V^2}{2Jc_p} \qquad (7)$$

where

t_s = static temp (°F)

t_t = total temp (°F)

J = mechanical equivalent of heat (778 ft-lb per Btu)

c_p has units of Btu per slug-°F.

Combining equations (6) and (7) and solving for the static temperature t_s gives the following:

$$t_s = t_{aw} - N_{rf}\left(\frac{V^2}{2Jc_p}\right)$$

$$t_s = 1{,}875 - 0.720\left(\frac{800^2}{2(778)\ (9.03)}\right)$$

$$t_s = 1{,}842°\text{F}.$$

The total temperature can now be calculated as follows:

$$t_t = t_s + \left(\frac{V^2}{2Jc_p}\right)$$

$$t_t = 1{,}842 + \left(\frac{800^2}{2(778)\ (9.03)}\right)$$

$$t_t = 1{,}888°\text{F}.$$

Summary

It is interesting to note the effect that the velocity of the gas stream has on the temperature indicated by the thermocouple. Since aerodynamic heating is proportional to V^2, the difference between indicated temperature and static temperature would increase if the velocity were decreased (equation 8). Also, since the heat transfer coefficient is proportional to $V^{0.5}$ (equation 5), a de-

Presented here is the solution to a typical problem assigned to students enrolled in advanced heat transfer course work at General Motors Institute. The solution requires the application of superimposed principles of heat transfer and fluid dynamics.

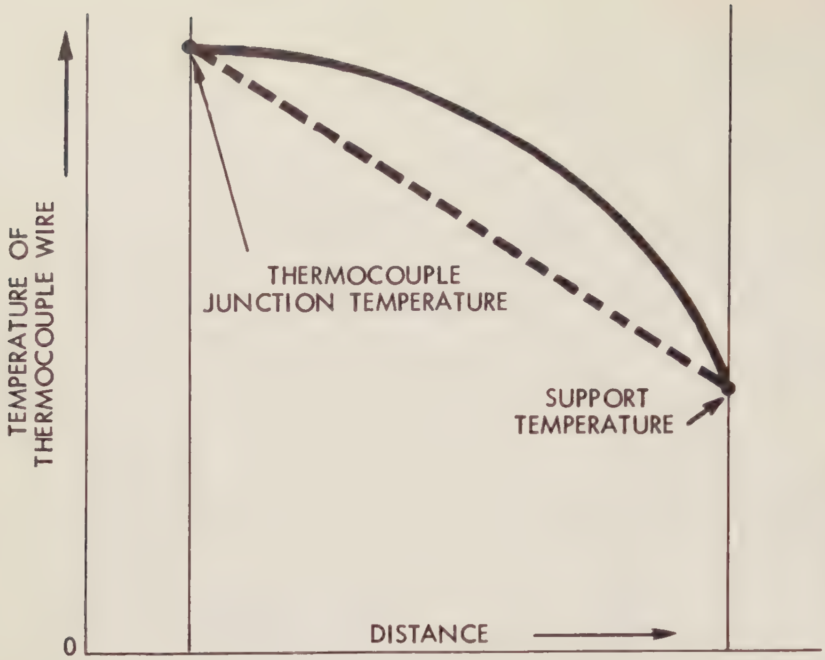


Fig. 2—These are typical curves representing the temperature distribution along the thermocouple wires from the thermocouple junction to support. The solid line represents the general shape of the actual temperature distribution. The broken line represents an assumed linear temperature distribution. Note that in comparison the linear plot, at the thermocouple junction, results in a larger temperature gradient than does the other plot.

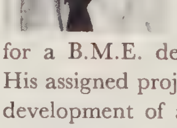
crease in velocity would lower the heat transfer coefficient, again resulting in a greater difference between indicated temperature and static temperature. Increasing the diameter of the thermocouple junction would also result in a greater difference in indicated and static temperatures, since the heat transfer coefficient is proportional to $D^{-0.5}$ (equation 5).

The radiation transfer component is the primary source of heat loss from the thermocouple junction at elevated temperatures because of the Stefan-Boltzmann fourth power law. As a result, it is important to note that for a given constant temperature difference between thermocouple junction and surrounding surfaces, the temperature level is the primary factor affecting the radiation transfer.

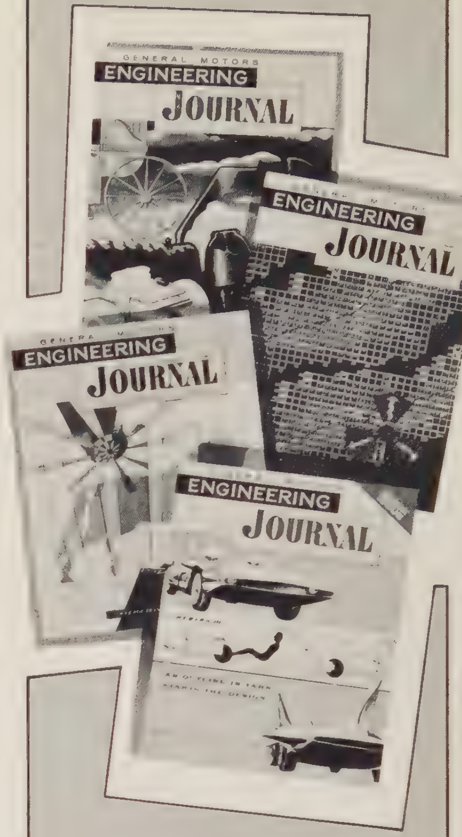
Bibliography

1. Jakob, M. and Hawkins, G., *Elements of Heat Transfer*, (New York: John Wiley and Sons, Inc., Third Edition, 1957), pp. 230-1.
2. Allen, S. and Hamm, J., "A Pyrometer for Measuring Total Temperature in Low Density Gas Streams," *Transactions of the A.S.M.E.*, Vol. 72, No. 6, 1950, p. 852.
3. McAdams, W. H., *Heat Transmission*, (New York: McGraw-Hill Book Co., Third Edition, 1954), pp. 309-16.

Student Contributor to the General Motors Institute Classroom Problem



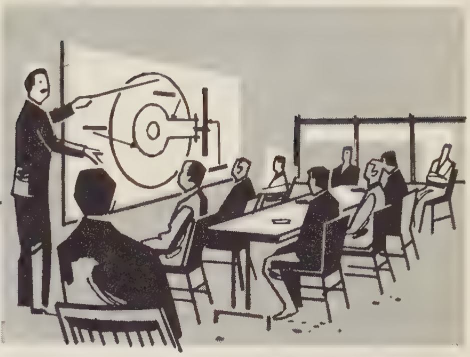
RODGER E. BLOOMFIELD is a junior engineer in the Gage Section, AC Spark Plug Division. He joined AC in 1954 as a G.M.I. co-op student. He presently is participating in the G.M.I. Fifth Year Program which will qualify him for a B.M.E. degree upon completion. His assigned project study deals with the development of a gas turbine temperature indicating system. During the Four-Year cooperative program his plant work assignments were in the fuel pump and air cleaner manufacturing areas, tool room, Silk Screen Section, Metallurgy Department, and Automotive Test Laboratories. Mr. Bloomfield is a member of the Society of Automotive Engineers and the American Institute of Electrical Engineers.



BACK COPIES of the four issues in Volume 5 of the General Motors Engineering Journal—whose covers are reproduced above—are available upon request.

As a service to readers, an index to the information presented in Volume 5 is published on page 60 of this issue. Indexes of previous volumes appeared in the following back issues: Volume 4, No. 4 (1957); Volume 3, No. 5 (1956); Volume 2, No. 6 (1955); Volume 1, No. 9 (1953-54). Figures in parentheses indicate the calendar year of publication for the issues of the volume. In most cases, extra copies of the issues in these volumes also are available.

Technical Presentations by GM Engineers



The technical presentation is another way in which information about current engineering and scientific developments in General Motors can be made available to the public. A listing of speaking appearances by General Motors engineers and scientists, such as that given below, usually includes the presentation of papers before professional societies, lecturing to college engineering classes or student societies, and speaking to civic or governmental organizations. Educators who wish assistance in obtaining the services of GM engineers and scientists to speak to student groups may write to the Educational Relations Section, Public Relations Staff, General Motors Technical Center, P. O. Box 177, North End Station, Detroit 2, Michigan.

GM personnel who have made recent technical presentations are as follows:

Automotive Engineering

David Eddy, section engineer, and **George Brown,** field contact engineer, Spark Plug Engineering Department, AC Spark Plug Division, before sales personnel and customers of Automotive, Inc., Tulsa, Oklahoma, May 20; title: Spark Plug Design and Servicing.

George Brown, before service and parts managers and garage owner customers of Puget Sound Auto Electric Company, Tacoma, Washington, May 22; title: AC Automotive Products and Servicing.

Glen R. Fitzgerald, director, Equipment Sales and Automotive Engineering Department, AC Spark Plug Division, before the National Automotive Parts Association, Omaha, Nebraska, May 27; title: What We May Expect in the Car of Tomorrow.

Richard F. Youngblood, project engineer, Product Engineering—Chassis Development Department, Oldsmobile Division, before the National Science Foundation, Michigan State University, East Lansing, June 25; title: Applied Mechanics in Air Suspension.

George W. Onksen, engineering head, Research and Development, Guide Lamp Division, before the Engineering Committee of the American Association of Motor Vehicle Administrators, Detroit, June 24; title: Headlamp Aiming.

F. Gibson Butler, group leader, design, Product Engineering—Motor Design Department, Oldsmobile Division, before the National Science Foundation, Michigan State University, East Lansing, June 25; title: Mathematics and Science in the Automotive Industry.

William S. Shade, staff engineer, Product Engineering Department, Diesel Equipment Division, before the Society of Automotive Testers, Rockford, Illinois, July 15; title: Hydraulic Valve Lifters.

Howard K. Gandelot, engineer in charge, Vehicle Safety Section, GM Engineering Staff, before the annual meeting of the Engineering Committee of the American Association of Motor Vehicle Administrators, Detroit, June 23, 25; titles: Automobile Theft Protection and Progress on License Plate Mountings, and before the Sixth Annual Meeting of the International Association of Auto Theft Investigators, Memphis, Tennessee, July 14; title: Theft Protection Efforts by the Automobile Manufacturers.

John F. Pribonic, section engineer, Advanced Automotive Design Department, Delco Products Division, before the Ta-Wa-Si, Dayton, Ohio, July 29; title: Automotive Air Suspension.

Harry D. Wright, Jr., car test supervisor, Car Test Engineering Department, Delco Products Division, before the Dealer Conference and Clinic Session, Shell Oil Company, Dayton, July 31; title: GM Air Suspension.

Max M. Roensch, assistant chief engineer, Experimental Tests Section, Engineering Department, Chevrolet Motor Division, before the S.A.E. west coast meeting, Los Angeles, August 12; title: Chevrolet's New Engineering Laboratory.

S.A.E. Summer Meeting Atlantic City, New Jersey

The following GM personnel made technical presentations at the S.A.E. annual summer meeting, Atlantic City, June 9 to 12: **Joseph B. Bidwell,** head, and **Roy S. Cataldo,** senior research engineer, Engineering Mechanics Department, GM Research Laboratories; title: How Will Future Motor Cars Affect Highway Construction. **Darl F. Caris,** engineer in charge, and **Edwin E. Nelson,** project engineer, Power Development Group, GM Engineering Staff; title: A New Look at High Compression Engines. **Louis C. Lundstrom,** director, GM Proving Ground; title: Safety Aspects of Vehicle-Road Relationships. **Philip O. Johnson,** engineer in charge, Physical Testing Laboratory, Fisher Body Division; title: Where Do We Find the Space for Seats? **W. M. Wiese,** senior research engineer, Fuels and Lubricants Department, GM Research Laboratories; title: If You Squeeze Them, Must They Scream?

Electrical Engineering

A. E. Manes, field service engineer, Service Department, Delco Radio Division, before members of the Long Island Radio and TV Guild, Mineola, New York, May 21; title: Transistor Theory and the Auto-Portable Radio.

Dr. R. F. Miller, senior research engineer, Semi-Conductor Department, Delco Radio Division, before the Institute on Industrial Applications of Transistors, Milwaukee, Wisconsin, June 3; title: Physical Properties of Transistor Operation.

Foundry

The following GM personnel made technical presentations at the American Foundrymen Society's Cleveland, Ohio, convention, May 19 to 23: **C. E. Fausel,** manufacturing superintendent, Manufacturing Department, Central Foundry Division; title: Your Foundry and Preventive Maintenance. **Richard A. Rabe,** project engineer, electronics, Process Development Section, GM Process Development Staff; title: High Temperature Properties of Shell Molds. **F. J. Webbere,** supervisor, Metallurgical Engineering Department, GM Research Laboratories and **H. A. Laforet,** Pontiac Motor Division; title: Duplexing Pays at Pontiac Foundry.

Karl Schwartzwalder, director of research, AC Spark Plug Division, before the Northern New York Section of the American Ceramic Society, Niagara Falls, June 6; title: Compression, Injection, and Hydrostatic Molding.

Carl F. Schaefer, supervisor, ceramic development and control, Research Department, AC Spark Plug Division, before the White Wares Symposium, New York State College of Ceramics, Alfred, June 25; title: Molding Alumina Ceramics.

Guided Missiles

Engineering personnel of AC Spark Plug Division's Milwaukee, Wisconsin, facility who made recent presentations on guided missiles and inertial guidance systems are as follows:

Dr. James Bell, engineering program director, Mace and Regulus missiles, before engineers of the Allis-Chalmers Corporation, Milwaukee, May 20; title: Missiles and Missile Guidance.

Donald L. Carpentier, Regulus missile project engineer, before the Milwaukee Junior Chamber of Commerce, May 21; title: The Missile and Its Challenge to Industry.

Jack H. Schmidt, engine control engineer, before the Air National Guard, Milwaukee, June 17; title: AC's Role in the Missile Field.

Ray A. Berg, assistant experimental engineer, before the Menominee Falls, Wisconsin, Rotary Club, June 18; title: General Comments on Inertial Guidance Systems.

Anthony J. Italiano, assistant technical director, Mace missile, before the Waukesha, Wisconsin, Kiwanis Club, June 19; title: Inertial Guidance, and before the Waukesha Optimist Club, July 16; title: Missiles and Inertial Guidance.

Kenneth Schlager, engineering supervisor, Regulus missile, before a graduate seminar in electrical engineering, Northwestern University, June 4; title: Inertial Navigation—A Practical Application of Precision Measurement and Control, before the Operations Research Society, Boston, May 15; title: An Application of Dynamic Programming to the Design of a Nonlinear Feedback Control System, and before the Milwaukee Optimist Club, July 28; title: America's Space Program.

Manufacturing

Halford D. Seeley, senior methods engineer, AC Spark Plug Division, before students of General Motors Institute, Flint, May 9; title: Methods Practices at AC Spark Plug Division.

Robert A. McCreedy, supervisor, Methods Engineering and Plant Layout Department, AC Spark Plug Division, before the Industrial Management Society, Grand Rapids, Michigan, May 9; title: Improving Profits Through Better Methods.

David Milne, supervisor, materials and processes, Production Engineering Section, GM Process Development Staff, before the Ontario, Canada, Resources Conference, May 27; title: Industrial Waste Control at General Motors.

Marshall McCuen, section chief, Advanced Design and Development, Allison Division, before the Indianapolis Scientech Club, June 5; title: Machining Without Chips.

J. M. Murphy, supervisor, major product line, Refrigerated Appliances Engineering Department, Frigidaire Division, before the Domestic Refrigerator Engineering Conference, Minneapolis, Minnesota, June 23; title: Mass Production Versus Serviceability.

Metallurgy

Harold A. Kahler, research chemist, Electrochemistry Department, GM Research Laboratories, before the American Electroplaters' Society, Cincinnati, May 19; title: Electroplating.

Raymond L. Mattson, assistant head, Engineering Mechanics Department, GM Research Laboratories, before the Detroit chapter of the American Society of Mechanical Engineers, June 17; title: Residual Stress, Cold Working, and Fatigue.

C. H. Ek, junior engineer, Special Problems Department, GM Research Laboratories, and **P. E. Hawkes,** no longer with GM, before the American Society of Testing Materials, Boston, June 24; title: A Machine for the Evaluation of High Temperature Alloys Under Combined Static and Dynamic Stresses.

Robert Davies, assistant head, Mechanical Development Department, GM Research Laboratories, before students of the University of Michigan, July 14; title: Compatability of Metal Pairs.

Miscellaneous Subjects

Robert M. Critchfield, vice president in charge of GM Process Development Staff, before students of the University of Detroit, May 15; title: Junior Achievement of Southeastern Michigan.

Donald W. Bain, director of drafting, Administrative Section, Engineering Department, Chevrolet Motor Division, before the Belsky Alumni Association, Detroit, May 21; title: Attributes of Executive Engineers.

Dr. Helen Bartlett, supervisor, Ceramic Research Department, AC Spark Plug Division, before the Kiwanis Club of Flushing, Michigan, May 22; title: Natural and Synthetic Gems.

Claude W. Willis, director of safety, Delco Products Division, over radio station WING, Dayton, June 9; title: Traffic Safety.

William R. Zakariasen, assistant superintendent, Maintenance and Construction Department, AC Spark Plug Division, before the Civitan Club, Flint, Michigan, June 16; and before the Burton Lions Club, Flint, July 15; title: Apprentice Training.

John Bryant, editor, *GM Engineering Journal*, before the Institute in Technical and Industrial Communications, Colorado State University, Fort Collins, Colorado, July 9; title: The Company Technical House Organ.

George P. Ransom, section engineer, Power Development Group, GM Engineering Staff, before the Mathematics Seminar for Teachers, University of

Richmond, Richmond, Virginia, July 16; title: Application of Mathematics to a Special Problem in the Design of the Axial Type Refrigeration Compressor.

Charles A. Nichols, technical assistant to the vice president in charge of GM Process Development Staff, before students of Cornell University, Ithaca, New York, July 17; title: Supervision of Professional and Technical Personnel—Aspects of Job Assignment and Utilization.

R. W. Chase, head, service engineering, Military and Electronics, AC Spark Plug Division, before the International Society of Grapho-Analysts, Flint, Michigan, July 19; title: Teamwork in Using Grapho-Analysis in Interviewing Engineering Job Applicants.

Charles A. Chayne, vice president in charge of GM Engineering Staff, before graduating students of General Motors Institute, August 8; title: Commencement is Just a Beginning.

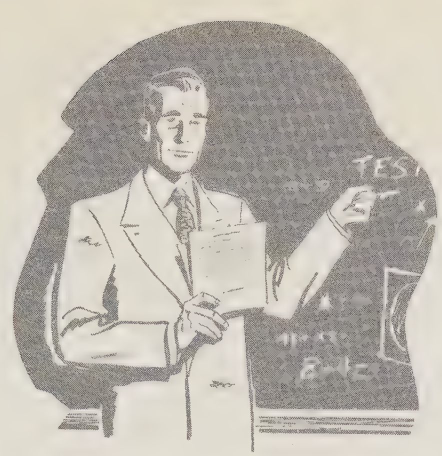
Research

James G. Roberts, research engineer, Engineering Mechanics Department, GM Research Laboratories, before the Colorado Springs chapter of the S.A.E., May 6; title: Correlation of Fatigue Strength and Residual Stresses Introduced by Shot-Peening.

Ralph A. Richardson, head, Administrative Engineering Department, GM Research Laboratories, before the Industrial Research Institute, Colorado Springs, May 19; title: Research Toward Specific Goals.

Kenneth A. Stonex, assistant director, GM Proving Ground, before the tenth Missouri Traffic Conference, Columbia, May 21; title: Research as Applied to Traffic and Transportation.

Robert Herman, assistant head, Basic Science Department, GM Research Laboratories, and **Robert J. Rubin,** National Bureau of Standards, before the Symposium on Molecular Structure and Spectroscopy, Columbus, Ohio, June 18; title: Theoretical Interpretation of Intensities of Vibration-Rotation Bands of Diatomic Molecules.



Carl E. Bleil, Donald D. Snyder, and Y.T. Sihvonen, senior research physicists, Physics Department, GM Research Laboratories, before the American Physical Society, Ithaca, New York, June 21; title: Conductivity Induced in CdS Crystals by 30 to 60 kev Electrons, and before the American Crystallographic Association, Milwaukee, Wisconsin, June 25; title: Electron Interactions in CdS Crystals.

H. L. Garabedian, assistant head, **Charles B. Leffert,** senior nuclear physicist, Nuclear Power Engineering, GM Research Laboratories, before the American Nuclear Society, Los Angeles, June 3; title: Kinetics of a Multi-Region Reactor Under Non-Separability Conditions.

Personnel of the Physics Department, GM Research Laboratories, who made technical presentations at the Electron Microscope Society of America meeting, Santa Monica, California, August 7 to 9 were as follows: **Stanely R. Rouze,** senior research physicist, and **F. Agnes Forster,** junior chemist; title: A Service Evaluation of Detergent Lubricating Oils Utilizing Electron Microscopy; **Robert L. Scott,** research chemist and **Robert V. Coleman,** senior research physicist; title: A Study of Deformation and Corrosion in Single Crystal Iron Whiskers by Electron Microscopy.

1958 General Motors Conference for Engineering and Science Educators

The Seventh GM Conference for Engineering and Science Educators was held in Detroit July 6 through July 22. Twenty-eight educator guests attended as representatives of colleges and universities from the United States and Canada. Co-hosts and chairmen of the Conference, designed to acquaint educators with GM product, production, and research activities, were: **Charles A. Chayne,** vice president in charge of Engineering Staff; **Robert M. Critchfield,** vice president in charge of Process Development Staff; and **Dr. Lawrence R. Hafstad,** vice president in charge of Research Laboratories. The Conference was co-ordinated by **Kenneth A. Meade,** director, Educational Relations Section, Public Relations Staff.

On July 7 an introductory general session was held in the morning at the GM Technical Center. GM personnel who made presentations at this session included: **William A. Collins,** assistant secretary, General Motors; **George A. Jacoby,** director, Personnel Relations, Personnel Staff; and **David F. Waggoner,** director, Salaried Personnel Placement, Personnel Staff. In the afternoon the educators visited the GM Styling Staff. **Harley J. Earl,** vice president in charge of Styling Staff, acted as host for the educator group. Styling Staff personnel who made presentations included: **Robert F. McLean,** executive in charge of styling product analysis and planning; **Donald D. Hoagg,** chief designer, Design Development Studio; **Stephan Habsburg,** assistant chief designer, Research "A" Studio; and **Dr. Peter Kyropoulos,** executive in charge of technical development.

On July 8 the educators visited the GM Engineering Staff. **Charles A. Chayne** discussed product engineering in GM. **Lyle A. Walsh,** manager, Engineering Staff Activities, discussed the Staff's activities. Other Engineering Staff personnel who made presentations included: **Maurice A. Thorne,** engineer in charge, Vehicle Development Group; **Gerhard C. Kuiper,** assistant engineer in charge, Passenger Car Section; **Lothrop M. Forbush,** assistant engineer in charge, Automotive Ordnance Section; **Von D. Polhemus,** engineer in charge, Structure and Suspension Development Group; **Stephen Kalmar,** executive assistant engineer in charge, Power Development Group; and **George R. Smith,** executive assistant engineer in charge, Transmission Development Group.

Dr. Lawrence R. Hafstad acted as host for the educator group when they visited the Research Laboratories on July 9. Chairman for the day's program was **John M. Campbell,** scientific director. Program speakers included **Dr. Alexander Somerville,** supervisor, Nuclear Physics and Isotope Applications; **Joseph B. Bidwell,** head, Engineering Mechanics

Department; and **Robert H. Kohr**, supervisor, Vehicle Dynamics. The educators were addressed at a luncheon by **A. G. De Lorenzo**, vice president in charge of Public Relations Staff.

The educators visited the Process Development Staff on July 10. **Robert M. Critchfield** acted as host for the group. Process Development Staff personnel who made presentations included: **Charles A. Nichols**, technical assistant to the vice president; **Harry D. Hall**, director, Process Development Section; **Robert D. McLandress**, director, Work Standards and Methods Engineering Section; **John Q. Holmes**, director, Production Engineering Section; **Peter L. Fucinari**, superintendent, Machine and Project Shop; **Lee M. Larson**, and **Carl W. Stenson**, senior contact engineers, **Earl T. Barringer**, chief designer; **Donald H. Johnson**, staff supervisor, Methods Engineering; and **Edward J. Conley**, staff engineer, Methods Engineering.

On July 11 the educators toured the GM Proving Ground facilities. **Louis C. Lundstrom**, director of the Proving Ground, acted as host. Proving Ground personnel who made presentations included: **A. H. Kelley**, head, Engineering Test Department; **Paul C. Skeels**, head, Experimental Engineering Department; and **David C. Apps**, head, Noise and Vibration Laboratory.

The educators visited General Motors Institute on July 12. **Guy R. Cowing**, president and director of G.M.I., acted as host and **Harold M. Dent**, administrative chairman, cooperative engineering program, served as moderator for a discussion period during which time presentations were given by: **Roy Totten**, instructor in charge of electrical control system courses; **Felix Mickus**, instructor in charge of thermodynamics courses; and **Harold Haskitt**, instructor in charge of speech courses. Luncheon speaker for the day's program was **John F. Gordon**, president, General Motors, and also chairman of the board of regents, G.M.I.

On July 21 the educators were addressed by **Louis C. Goad**, executive vice president, General Motors.

The educators met for the closing general session of the Conference at the GM Technical Center on July 22. Dr. Gordon B. Carson, vice president, The Ohio State University, presented "Some Thoughts on Science and Engineering Education," followed by a discussion period and the Conference adjournment.

Solution to the Previous Problem:

Determine the Effect of Changes in Ambient Temperature on an Automotive Air Suspension System

The designer of an automotive air suspension system is concerned not only with design problems dealing with the sizing of air springs for proper spring rate, frequency, and deflection but also with additional design problems created by temperature, humidity, and altitude. This is the solution to the problem presented in the July-August-September issue of the GENERAL MOTORS ENGINEERING JOURNAL in which one of these additional design problems was considered—namely, the effect a change in ambient temperature has on the pressure of the air tank component of the suspension system and the air mass transferred when the car is parked over a period of time. A decrease in ambient temperature results in an air tank pressure of *250 psia*. The mass of air which must be transferred to maintain normal ride height when the temperature decreases is *0.0128 lb*. With increased temperature the air tank pressure increases to *275 psia*. The mass of air which must be transferred to maintain normal ride height with increased temperature is *0.0128 lb*.

THE first step in the solution of the problem is to analyze the data to determine whether the following limitations of the fundamental gas relations satisfy the conditions of the problem:

- The working medium must remain a gas throughout the process
- The temperature of the gas must be well above its critical temperature
- Temperatures must be expressed as absolute temperatures (°Rankine)
- Pressures must be expressed as absolute pressures (psia)
- The mass of gas between two or more operational points must be constant or accountable.

Since the temperature of the ambient air, as stated in the problem, is more than twice the critical temperature of either nitrogen or oxygen, the general gas relation is satisfactory. The results obtained from the general gas relation will be within one per cent if condensation or evaporation does not occur. Since the major portion of water vapor condensation occurs in the air tank while the compressor is in operation, it can be assumed that for the non-operating automobile the water vapor in the compressed air is superheated.

The problem stated that the ambient temperature decreased from 80°F to 60°F. For this condition, the pressures in both the air tank and the air springs decrease (Fig. 1). But, as the air springs lower the car body, the height control valves open and allow sufficient air into the springs to return the body to its normal height. The valves then shut off the air supply when the body reaches its normal height position. Since the air compressor is not in operation, the air supplied to the springs comes from the air tank. No air is released from the system. The mass of air in the system at 80°F, therefore, is equal to the mass of air in the system at 60°F, or

$$M_{t80} + M_{s80} = M_{t60} + M_{s60} \qquad (1)$$

where

M_{t80} = mass of air in tank at 80°F (lb)

M_{s80} = mass of air in springs at 80°F (lb)

M_{t60} = mass of air in tank at 60°F (lb)

M_{s60} = mass of air in springs at 60°F (lb).

From the following characteristic equation of a perfect gas

$$PV = MRT,$$

the mass of air M will be equal to

$$M = \frac{PV}{RT} \qquad (2)$$

By ERIC DIETRICH
Buick Motor Division
and LEONARD B. WOCHOLSKI
General Motors Institute

Basic gas relationships applied to obtain desired information

where

M = mass of air (lb)

P = absolute pressure (psia). Atmospheric pressure is taken as being equal to 15 psi

V = volume of air (cu ft)

T = absolute temperature of air (°Rankine)

R = gas constant for air

$R = 0.37\,\dfrac{(\text{psia})\ (\text{ft}^3)}{(\text{lb})\ (°\text{R})}$.

Substituting the value for M from equation (2) into equation (1) gives the following:

$$\frac{(P_{t80})\ (V_t)}{(R)\ (T_{80})}+\frac{(P_{s80})\ (V_s)}{(R)\ (T_{80})}=\frac{(P_{t60})\ (V_t)}{(R)\ (T_{60})}+\frac{(P_{s60})\ (V_s)}{(R)\ (T_{60})}$$

where

P_{t80} = absolute pressure of air tank at 80°F (psia)

V_t = volume of air tank (cu in. or cu ft)

T_{80} = absolute temperature at 80°F (°R)

P_{s80} = absolute pressure of air springs at 80°F (psia)

V_s = volume of air springs (cu in. or cu ft)

P_{t60} = absolute pressure of air tank at 60°F (psia)

T_{60} = absolute temperature at 60°F (°R)

P_{s60} = absolute pressure of air springs at 60°F (psia).

Since the load W does not change, the air spring pressure remains constant, or

$$P_{s80} = P_{s60}.$$

The absolute pressure in the air tank, therefore, can be calculated as follows:

$$P_{t60} = P_{t80}\left(\frac{T_{60}}{T_{80}}\right) + P_{s80}\left(\frac{V_s}{V_t}\right)\left(\frac{T_{60}}{T_{80}}\right) - P_{s60}\left(\frac{V_s}{V_t}\right). \tag{3}$$

Substituting known values into equation (3) and solving for the absolute pressure of the air tank at 60°F gives

$$P_{t60} = 250 \text{ psia}.$$

The gage pressure for the air tank at 60°F will be

$$250 \text{ psia} - 15 \text{ psi} = 235 \text{ psig}.$$

The mass of air transferred from the air tank to the air springs will be equal to the difference between the mass of air in the tank at 80°F and the mass of air in the tank at 60°F, or

$$M = M_{t80} - M_{t60}$$

$$M = \frac{(P_{t80})\ (V_t)}{(R)\ (T_{80})} - \frac{(P_{t60})\ (V_t)}{(R)\ (T_{60})} \tag{4}$$

where

the units of V_t must be cu ft,
if $R = 0.37\,\dfrac{(\text{psia})\ (\text{ft}^3)}{(\text{lb})\ (°\text{R})}$.

Substituting known values into equation (4) will give the air mass M transferred to the air springs to maintain normal ride height at 60°F. This value is equal to 0.0128 lb.

The increased ambient air temperature conditions were stated in the problem as 80°F for the initial air temperature and 100°F for the final air temperature. Under these conditions the air pressure in the air springs increases causing the car body to rise above its normal ride height. This action, in turn, causes the height control valves to open and discharge a sufficient amount of air to lower the body back to its normal height. The height control valve is closed whenever the body is at the normal ride height.

The mass of air transferred is entirely from the air springs. The mass of air in the springs, therefore, at 80°F (M_{s80}) is equal to the mass of air in the springs at 100°F (M_{s100}) plus the mass of air discharged from the springs (M_d) to maintain normal ride height, or

$$M_{s80} = M_{s100} + M_d. \tag{5}$$

The mass of air discharged from the springs is discharged to the atmosphere. This mass can be calculated by rearranging equation (5) as follows:

$$M_d = M_{s80} - M_{s100}$$

$$M_d = \frac{(P_{s80})\ (V_s)}{(R)\ (T_{80})} - \frac{(P_{s100})\ (V_s)}{(R)\ (T_{100})} \tag{6}$$

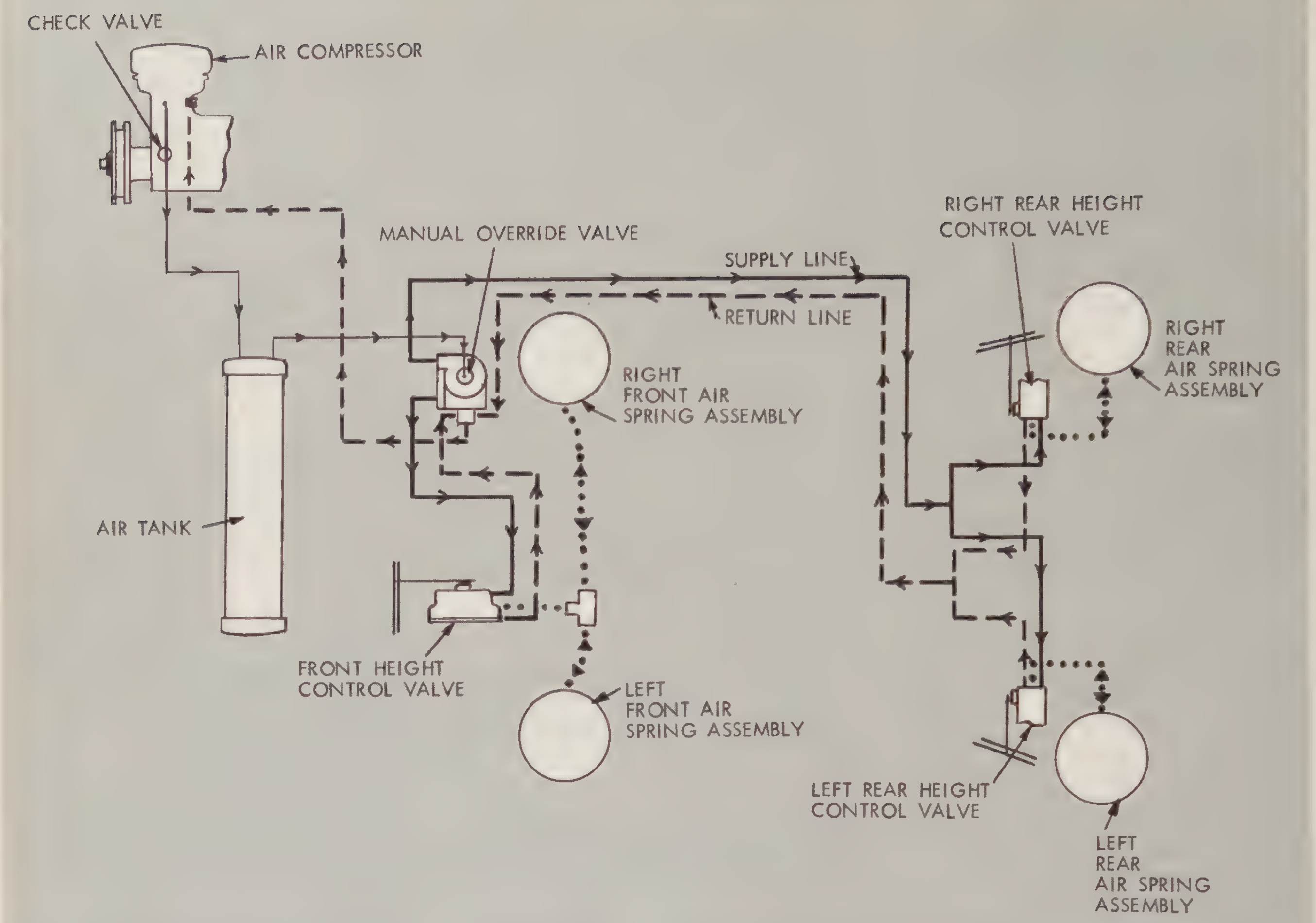

CHECK VALVE
AIR COMPRESSOR
MANUAL OVERRIDE VALVE
SUPPLY LINE
RETURN LINE
RIGHT REAR HEIGHT CONTROL VALVE
RIGHT REAR AIR SPRING ASSEMBLY
RIGHT FRONT AIR SPRING ASSEMBLY
AIR TANK
FRONT HEIGHT CONTROL VALVE
LEFT FRONT AIR SPRING ASSEMBLY
LEFT REAR HEIGHT CONTROL VALVE
LEFT REAR AIR SPRING ASSEMBLY

Fig. 1—Shown here is a schematic arrangement of a typical automotive air suspension system. The engine drives the air compressor which delivers air to the air tank at a pressure of from 250 psig to 280 psig. The air tank acts as storage for the compressed air and also as a trap for the condensed water vapor and oil that passes through the compressor. Air from the tank passes through 0.1875-in. diameter steel tubing to the manual override valve which normally reduces and regulates the 250 psig air to a constant pressure of 125 psig. This air is then passed to the front height control valve and the two rear height control valves. These three height control valves work together to keep the car at a normal height regardless of the load or ambient conditions and they also control the flow of air into or out of the air spring. Air is supplied to the air springs, by the height control valves, for increased load or decreased ambient temperatures. Air is removed from the air springs for decreased load, increased ambient temperature, or any other ambient condition which tends to increase the volume of the air spring. Air is discharged from the air springs through the height control valve. This discharged air is returned to the suction side of the compressor where it may either be used over again or discharged to the atmosphere.

where

V_s must be in cu ft

if $R = 0.37 \dfrac{(\text{psia})\ (\text{ft}^3)}{(\text{lb})\ (^\circ\text{R})}$.

Substituting known values into equation (6) will give the mass of air discharged from the springs to the atmosphere in order to maintain normal ride height. This value is 0.0128 lb.

For the case where there is an increase in ambient air temperature conditions, no air will be transferred from the air tank. The increase in ambient air temperature, therefore, will increase the pressure in the air tank according to the following constant volume relationship:

$$\frac{P_{t80}}{T_{80}} = \frac{P_{t100}}{T_{100}}$$

or

$$P_{t100} = P_{t80}\left(\frac{T_{100}}{T_{80}}\right) \qquad (7)$$

where

P_{t100} = absolute pressure in the air tank (psia)

T_{100} = absolute temperature of the air at 100°F (°R).

Substituting known values into equation (7) gives the value for the absolute pressure in the air tank at 100°F as 275 psia. The gage pressure for the air tank at this temperature is 260 psig.

Staff Members of the GENERAL MOTORS ENGINEERING JOURNAL

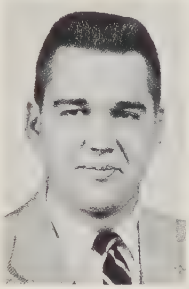
In response to occasional requests about the people who serve on the staff of the JOURNAL, the last issue included brief biographical sketches of the editorial staff. In this issue, information is presented about the art and production staff. It may be of interest to note that—in addition to their Journal responsibilities—the individuals mentioned below perform similar duties relating to other publication projects in the GM Public Relations Staff. Among the different types of publications are many which have been prepared for use by educators. Typical examples are: (a) several vocational guidance booklets such as the recently revised "Can I Be a Scientist or Engineer?" (b) a series of secondary school science resource units called "Science at Work in General Motors," and (c) various booklets on technical subjects such as "Power Primer," "How the Wheels Revolve," and "ABC's of Hand Tools."

ERNEST W. SCANES,

Art Director, studied art as a scholarship holder at the John P. Wicker School of Fine Arts (1928-1929). He also attended the Society of Arts and Crafts School (1930-1931) and received a fellowship to Cranbrook Art Academy (1939-1941). His GM experience includes various art assignments in the Research Laboratories and the Central Office, and, since 1945, in the Public Relations Staff. He paints and exhibits regularly.

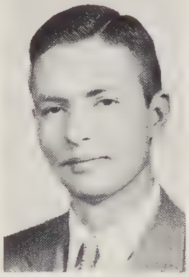
JOHN B. TABB,

Assistant Art Director, studied engineering at Purdue University (1941-1943) and received electronics training at Michigan State University (1943-1945) while in the U.S. Air Force. His art training was obtained at Meinzinger Art School. He taught art at Meinzinger and at Marygrove College in Detroit before joining GM in 1953. Active in several art societies, he paints and exhibits regularly.



RICHARD P. RENIUS,

Artist, attended Michigan State University (1949-1950) and Valparaiso University (1950-1951) and the Society of Arts and Crafts School. He joined General Motors in his present position in 1956 after six years of experience in visual aids, technical illustration, graphic arts, layout and design with automotive engineering departments, advertising agencies, and technical publications.



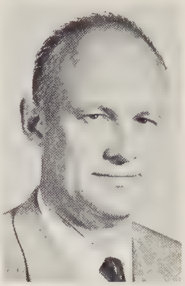
JOHN A. DICKEY,

Artist, was graduated from Wayne State University in 1955 with a B.F.A. degree, majoring in art. Following graduation he joined the GM Public Relations Staff handling art assignments for the JOURNAL and other projects. He began a leave of absence for military service in 1956 and recently returned to his present position.

DONALD O. BRYSON,

Production Manager, has 38 years of experience in various phases of the graphic arts. He joined GM in 1941. Previously, he served in commercial printing plants as typographer-printer, foreman, and general superintendent. He is a member of 23 years' standing in the International Club of Printing House Craftsmen. For the JOURNAL, he handles such matters as typography, paper selection, and contacts with suppliers.

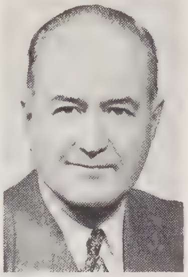
Contributors to Oct.-Nov.-Dec. 1958 Issue of ENGINEERING JOURNAL



ERIC R. DIETRICH,

co-contributor of the problem "Determine the Effect of Changes in Ambient Temperature on an Automotive Air Suspension System," and the solution appearing in this issue, is a senior project engineer in the Engineering Department of Buick Motor Division, Flint. His present responsibilities include the design and development of future air suspension systems and related components.

Mr. Dietrich joined GM in 1947 as a senior designer with Buick. In 1955 he was assigned to the chassis division to design and develop an air suspension system for the Buick passenger car. He is a member of the Society of Automotive Engineers Air Spring Sub-committee.

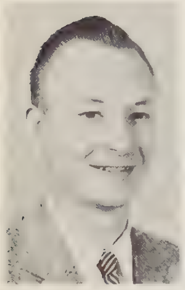


FRED E. JONES,

contributor of "The Filing and Maintenance of Patents in Foreign Countries," and coordinator of this issue's "Notes About Inventions and Inventors," is a patent attorney in the General Motors Patent Section, Detroit Office. He joined General Motors nearly 40 years ago as patent draftsman in the Detroit office. He was advanced to the position of patent attorney in 1929. Included in his duties are the filing and maintenance of General Motors patents in foreign countries, of which he writes in this issue.

Mr. Jones earned the Bachelor of Laws degree from Detroit College of Law. He is a member of the State Bar of Michigan. His society memberships include the American Patent Law Association and the Michigan Patent Law Association.

ALEX H. JOYCE,



co-contributor of "Dynamic Air Gaging Reduces Cycle Time in Precision Measurements," is a project engineer in the Engineering Department, Process Development Section, GM Process Development Staff. His present responsibilities include developing a new metallurgical process involving selective heat transfer.

Mr. Joyce joined the Process Development Staff in January 1956. His previous projects have included the development of an automatic turbine blade contour checking machine, an annular fluid dispenser for casting plastisol air filter flanges, and an automatic compression molding machine for small plastic parts. Prior to 1956, Mr. Joyce was a process development engineer and supervisor of the Davenport Works Laboratory of the Canadian General Electric Company.

Mr. Joyce received the bachelor of applied science in chemical engineering degree from the University of Toronto in 1949.

LOUIS H. RAVITCH,

contributor of "Some Applications of Stress Analysis Techniques in Improving Casting Designs," is stress analyst in the Experimental and Development Department, Central Foundry Division, Saginaw, Michigan. Mr. Ravitch's experience at Central Foundry includes the design of pattern equipment and recommending improvements in casting design as well as the stress analysis work of which he writes in this issue. He joined the Division in 1947 as a process engineer. He assumed his present position in 1956.

Mr. Ravitch took undergraduate engineering work at the University of Michigan from 1942 to 1945, and studied strain gage techniques in a summer program at Massachusetts Institute of Technology in 1956. He is a member of the Society for Experimental Stress Analysis.

Mr. Ravitch has played a prominent part in the establishment of Central Foundry Division's Stress Laboratory and the selection and installation of equipment as well as correlating the test work on casting designs. His presentation on the use of stress analysis has been one of the features of a series of 14 Casting Design Conferences conducted by Central Foundry Division for the information of casting users and educators interested in foundry technology.

> **Contributors' backgrounds vary greatly in detail but each has achieved a technical responsibility in the field in which he writes.**

KENNETH E. RUFF,

co-contributor of "Dynamic Air Gaging Reduces Cycle Time in Precision Measurements," is a senior project engineer in the Electronics Department of the General Motors Process Development Staff, located at the GM Technical Center.

Mr. Ruff joined the Process Development Staff in 1956 as a project engineer. He became a senior project engineer in 1958, and currently supervises the analysis and instrumentation group of the Electronics Department. Mr. Ruff received the M.A. degree from Wayne State University in 1956.

Mr. Ruff's previous projects have included the solution by analytical techniques of problems in mechanics, heat transfer, electronics, and statistical analysis.

PAUL C. SKEELS,

contributor of "Simple Two-Wheel Trailer Measures Road Surface Coefficient of Friction," is head of the Experimental Engineering Department at the GM Proving Ground, Milford, Michigan. Mr. Skeels joined the Proving Ground in 1940 as a laboratory technician. He was made head of the Electronics Department in 1946 and assumed his present position in 1953.

Mr. Skeels is currently responsible for the direction of developmental work on special test instrumentation, the conduct of special tests, and the development of new test techniques at the Proving Ground. These special test techniques are applied to such areas as car-to-car and car-to-fixed object crash tests; high-speed motion picture photography; headlight testing; and electronic instrument development.

Mr. Skeels is a member of the Society of Automotive Engineers and the Instrument Society of America. He also is a member of the S.A.E. Brake Subcommittee; the I.S.A. Transportation Committee; and Subcommittee E, International Skid Prevention Conference.

RALPH M. STALLARD,

co-contributor of "The Application of Dielectric Heat in Body Interior Trim Fabrication," is senior production engineer in the Process Development Laboratory, Production Engineering Activity, Fisher Body Division.

Mr. Stallard joined GM in 1949 as a junior process engineer at the Fisher Body Division, Process Development Plant, Detroit. His early projects included the testing of welding transformers and the development of electromagnetic devices. He was then assigned to the development of dielectric embossing methods and equipment for automotive trim applications. While working on these developments he was promoted to process engineer and then senior production engineer. In 1954 he was made group engineer responsible for the technical development of dielectric heating and other advanced production methods for automotive trim fabrication.

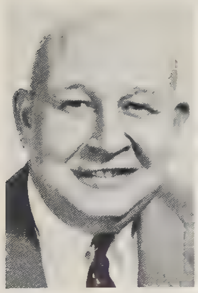
Mr. Stallard received a B.S.E.E. degree from the Michigan College of Mining and Technology in 1949. He is a member of the Engineering Society of Detroit.

WESLEY J. TRATHEN,

contributor of "The Application of a Magnus Effect Flow Meter as a Laboratory Teaching Device," is faculty member in charge of fluid mechanics courses at General Motors Institute. In addition to teaching courses in fluid mechanics and fundamentals of engineering design, he also is responsible for the development of fluid mechanics laboratory equipment.

The Michigan College of Mining and Technology granted Mr. Trathen a B.S.M.E. and M.E. degree in 1932 and a M.S. degree in 1933. Two years later he joined G.M.I. as both co-operative student and faculty member. Upon completion of his training in 1936, he received a certificate in Diesel engineering and was then employed as a full time faculty member until 1938. He then went to Electro-Motive Division and in October of 1939 returned to G.M.I. and taught courses in mathematics and mechanical engineering until assuming his present position.

Mr. Trathen is a registered mechanical engineer in the state of Michigan.



LEONARD B. WOCHOLSKI,

co-contributor of the problem "Determine the Effect of Changes in Ambient Temperature on An Automotive Air Suspension System," and the solution appearing in this issue, is associate chairman of the Science Department at General Motors Institute. He has supervision over courses taught within this Department's physics, electrical, and heat transfer sections.

Mr. Wocholski joined G.M.I. in 1934 as an instructor in heat transfer and physics. He was later promoted to assistant chairman of the Science Department and assumed his present position in 1956. Before joining GM, Mr. Wocholski was an air conditioning, ventilation, and refrigeration design engineer and also chief engineer of a power plant. He received a B.S.M.E. degree in 1932 and a M.S.M.E. degree in 1933, both from Purdue University.

Mr. Wocholski is a member of Pi Tau Sigma and Sigma Xi, honorary societies, and also the American Society for Engineering Education, the American Society of Physics Teachers, the American Society for the Advancement of Science, and the American Society of Refrigeration Engineers.

KENNETH W. WOODFIELD,

faculty member in charge for the typical General Motors Institute classroom problem "Determine the Static and Total Temperatures of a High Temperature, High Velocity Gas Stream" and the solution appearing in this issue, is head of the Heat Transfer Section of the Science Department at G.M.I. He is responsible for courses covering basic and advanced heat transfer, heat transfer design, industrial heating, and air conditioning.

Mr. Woodfield joined General Motors in 1943 as a G.M.I. co-op student sponsored by GMC Truck and Coach Division. His studies were then interrupted by service in the Navy, and resumed in 1946. He was granted a B.M.E. degree in 1950, and then joined the Science Department faculty. He assumed his present position in 1957.

Mr. Woodfield did graduate work at the University of Michigan, and received a M.S. degree in physics in 1956. He is a member of the Michigan Engineering Society and the American Society for Engineering Education.

PATON M. ZIMMERMAN,

co-contributor of "The Application of Dielectric Heat in Body Interior Trim Fabrication," is senior production engineer in the Process Development Department, Production Engineering Activity, Fisher Body Division.

Mr. Zimmerman joined GM in 1942 as a General Motors Institute co-op student with the Eastern Aircraft Division. He received a bachelor's degree in mechanical engineering in 1947 from G.M.I., where he was a member of Alpha Tau Iota honorary society.

In 1946, he transferred to the Buick-Oldsmobile-Pontiac Assembly Division as a junior process engineer. He was transferred to the same position in Fisher Body in 1947, where subsequent promotions have included process engineer, senior process engineer, and senior production engineer.

Mr. Zimmerman currently is serving as director of a program for the use of industrial closed-circuit television, consultant on industrial noise problems, and technical editor of the Process Development Department.

Mr. Zimmerman, who is a frequent contributor to scientific literature, is a member of the Society of Technical Writers and Editors and the General Motors Committee on Engineering Computation.

GENERAL MOTORS
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index to volume 5

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In This Index—

Page References

The number preceding the dash denotes the issue number. The number following the dash indicates the page number in that issue. Issue numbers are as follows:

1 January-February-March 1958
2 April-May-June 1958
3 July-August-September 1958
4 October-November-December 1958

Example: 2-38 means page 38 of the April-May-June 1958 issue.

Abbreviations

(CP) Classroom problem
(Ed) Editorial—always appears on unnumbered inside front cover
(P) Portrait

A

ACOUSTICS
  Design of automobile radios requires the solution of unique acoustical problems - - - - - 2—38

AIR CONDITIONING SYSTEM COMPONENTS
  Digital computer aids in design of refrigerant evaporator and centrifugal blower and scroll assembly - - - - - - - - - - 3—17

AIR-FUEL RATIO
  See Classroom Problems

AIR GAGES
  See Gages—Air

AIR SUSPENSION SYSTEMS
  See Suspension Systems—Automotive

AMANN, CHARLES A.
  Biography (P) - - - - - - - - - 2—61
  Testing Paper Gas Turbines: Applying the Digital Computer to Engine Design - - - - - 2—6

ANALYSIS
  See Stress Analysis, Vibration Analysis

AUTOMATIC CONTROL
  See Servomechanisms, Steering Equipment—Automobile

AUTOMOBILE DESIGN
  See also Product Design, Testing
  Firebird III: a new concept in automotive engineering - - - 4—33

AUTOMOBILE ENGINES
  See Engine Design

AUTOMOBILE HEADLAMPS
  See Headlamps—Automotive

AUTOMOBILE RADIO DESIGN
  A discussion of some factors affecting reproduced radio sound in an automobile - - - - - - - - 2—38

AUTOMOBILE STEERING SYSTEMS
  See Steering Equipment—Automobile

B

BALANCING FIXTURES
  Manufacturing engineers develop new fixture to statically balance speedometer drag cups - - - - 1—36

BARNARD, RICHARD P.
  A Dilemma of Group Research—Determination of Inventorship 1—40
  Biography (P) - - - - - - - - - 1—58

BEAMS—COMPOSITE
  See Classroom Problems

BENDA, EDWARD K.
  Biography (P) - - - - - - - 1—58
  Electronic Control System Provides Indoor Proving Grounds for Transmissions - - - - - - 1—25

BENDER, EDWARD A.
  Biography (P) - - - - - - - - - 2—62
  Magnetic Test Accurately Compares Heat Extraction Properties of Quenching Media - - - 2—31

BLOOMFIELD, RODGER
  Biography (P) - - - - - - - - - 4—50
  Determine the Static and Total Temperatures of a High Temperature, High Velocity Gas Stream (CP)
    Problem - - - - - - - - - - 3—53
    Solution - - - - - - - - - - 4—47

BORCHERTS, ROBERT H.
  Biography (P) - - - - - - - - - 1—55
  Determine the Center of Gravity Location of a Passenger Car (CP)
    Problem - - - - - (Vol. 4) 4—45
    Solution - - - - - - - - - - 1—54

BACK ISSUES: Readers may obtain back issues of the GENERAL MOTORS ENGINEERING JOURNAL as available upon request to the Educational Relations Section.

BRINKERHOFF, DONALD E.
A Discussion of Some Factors Affecting Reproduced Radio Sound in an Automobile - - - 2—38
Biography (P) - - - - - - - - - 2—62

BRYANT, JOHN
Biography (P) - - - - - - - - - 3—60

BRYSON, DONALD O.
Biography (P) - - - - - - - - - 4—57

C

CASTINGS
Casting designs improved by applying stress analysis techniques 4—22

CHASSEE, HAROLD E.
Biography (P) - - - - - - - - - 2—62
The Design and Development of a Synchronous Firing Device for Twin 40mm Guns - - - - - 2—26

CHEMICAL ANALYSIS
See Classroom Problems, Waste Disposal

CLASSROOM PROBLEMS
See also Mathematics, Product Design
Determine the center of gravity location on a passenger car
  Problem - - - - - (Vol. 4) 4—45
  Solution - - - - - - - - - 1—54
Determine the effect of changes in ambient temperature on an automotive air suspension system
  Problem - - - - - - - - - 3—55
  Solution - - - - - - - - - 4—54
Determine the operating characteristics of a drive and control system for a transfer mechanism
  Problem - - - - - (Vol. 4) 4—49
  Solution - - - - - - - - - 1—48
Determine the static and total temperatures of a high temperature, high velocity gas stream
  Problem - - - - - - - - - 3—53
  Solution - - - - - - - - - 4—47
Determine the thermal deflection of a composite beam
  Problem - - - - - - - - - 2—53
  Solution - - - - - - - - - 3—58
Determine the unbalanced forces in a converted V-8 engine
  Problem - - - - - - - - - 1—56
  Solution - - - - - - - - - 2—54
Develop an alignment chart to find air-fuel ratios from an Orsat analysis for the combustion of C_8H_{18}
  Problem - - - - - - - - - 4—46
  Solution - - - - - - - - (Vol. 6)
Find the resistance values of a thermocouple-potentiometer circuit
  Problem - - - - - - - - - 2—58
  Solution - - - - - - - - - 3—56

COLEMAN, DR. ROBERT V.
Biography (P) - - - - - - - - - 3—61
Single Crystal Iron Whiskers Adapted to the Basic Study of Magnetic Domains - - - - - 3—34

COMPUTERS—ANALOG
See also Mathematics
Analog computers used in the design and development of control systems at AC Spark Plug Division - - - (inside back cover)—1

COMPUTERS—DIGITAL
A basic discussion of digital computers - - - - - - - - - - - 2—2
Digital computer applied to the design of air conditioning system components - - - - - - 3—38
Digital computer applied to the design of a gas turbine engine - 2—6

CONTROL SYSTEMS
Electro-magnetic control system steers vehicles automatically - 2—60
Electronic control system provides indoor proving ground for transmission testing - - - 1—25
Firebird III embodies new concepts in automatic control systems for automotive vehicle application - - - - - - - - 4—33
Servo-controlled, electrohydraulic simulator used to test vehicle suspension systems - - - - - 2—12

CURTICE, HARLOW H.
Foreword to: Fifty Years of Progress—Some Typical Contributions of General Motors Engineers and Scientists - - - - - 3—2

D

DEFLECTION
See Classroom Problems

DELORENZO, ANTHONY
On Science, Engineering, and Public Relations (Ed) (P) - - 3

DICKEY, JOHN
Biography (P) - - - - - - - - - 4—57

DIELECTRIC HEATING
The application of dielectric heat to body interior trim fabrication 4—7

DIETRICH, ERIC
Biography (P) - - - - - - - - - 4—57
Determine the Effect of Changes in Ambient Temperature on an Automotive Air Suspension System (CP)
  Problem - - - - - - - - - 3—55
  Solution - - - - - - - - - 4—54

DIGITAL COMPUTERS
See Computers—Digital

DYNAMOMETERS
See Testing

E

ELECTRODYNAMIC VIBRATOR
See Vibration Analysis

ELECTRO-MECHANICAL ANALOGY
See Vibration Analysis

EMBOSSING
See Dielectric Heating

EMMONS, WILLIARD O.
Biography (P) - - - - - - - - - 3—61
The Design and Development of a Blood Heat Exchanger for Open Heart Surgery - - - - - - - 3—38

ENGINE DESIGN
Chevrolet W engine represents a new concept in V-8 engine design - - - - - - - - - - - 3—10
Determine the unbalanced forces in a converted V-8 engine (CP)
  Problem - - - - - - - - - - 1—56
  Solution - - - - - - - - - - 2—54
Gas turbine engine designed with the aid of a digital computer - 2—6
The principles of the free-piston engine and some of its applications - - - - - - - - - - - 3—22

ENGINEERING
On the Challenge of the Space Age (Ed) by C. R. Osborn (P) 4
On Science, Engineering, and Public Relations (Ed) by Anthony DeLorenzo (P) - - - 3
Research as applied to traffic engineering - - - - - - - - 1—12
Studies of radioisotopes show promise for present and future applications in engineering - - 2—17

ENGINEERING ASSIGNMENT IN GM
Inside back cover
Analog computer used in the design and development of control systems - - - - - - - 1
Electrohydraulic servomechanism requires evaluation tests as one part of developmental program 2
Methods laboratory utilized to train operators during preproduction pilot run - - - - 3
Special machines provide opportunities for versatile young engineers - - - - - - - - - 4

ENGINEERING—CAREERS
Why a Technical Education? (Ed) by John F. Gordon (P) - - - 1

ENGINEERS
Some typical contributions of GM engineers during the past 50 years - - - - - - - - - - - 3—2
Speaking engagements of General Motors engineers - - - 1—51, 2—50, 3—48, 4—51

EXPERIMENTAL CARS
Firebird III, experimental gas turbine powered car, has new concepts in automotive engineering - - - - - - - - - - 4—33

EZOP, EUGENE
Biography - - - (inside back cover)—4

F

FIREBIRD III
See Experimental Cars

FISHER, RUSSELL V.
Biography (P) - - - - - - - - - 3—61
Find the Resistance Values of a Thermocouple-Potentiometer Circuit (CP)
  Problem - - - - - - - - - 2—58
  Solution - - - - - - - - - 3—56

FLUID MECHANICS
The application of a Magnus effect flow meter as a fluid mechanics laboratory teaching device - - - - - - - - - - - 4—12

FLYNN, GREGORY, JR.
Biography (P) - - - - - - - - - 3—61
Some Principles and Applications of the Free-Piston Engine - - - 3—22

FOUNDRY PRACTICE
Stress analysis techniques improve casting designs - - - - - - - 4—22

FREE-PISTON ENGINE
See Engine Design

(CP) Classroom problem; (Ed) Editorial—always appears on unnumbered inside front cover; (P) Portrait

Friction—Road Surface
- Simple two-wheel trailer measures road surface coefficient of friction 4—14

G

Gages—Air
- Dynamic air gaging reduces cycle time in precision measurements 4—28

Gages—Radioisotope
- Radioisotope measuring gage applied to the production of storage batteries 1—34

Galin, Robert
- Biography (P) 3—62
- The Application of a Digital Computer to the Design of Air Conditioning System Components 3—17

Gallant, Raymond A.
- Biography (P) 1—58
- Electronic Control System Provides Indoor Proving Grounds for Transmissions 1—25

Gambill, Charles C.
- An Application of Radioisotope Wear Study Techniques 2—21
- Biography (P) 2—62

Gas Turbine Engines
- Firebird III has improved gas turbine engine 4—33
- Gas turbine engine designed with the aid of a digital computer 2—6

Gelenius, Robert
- Biography (inside back cover) 1

General Motors
- A review of some typical contributions made by GM engineers and scientists to technological progress since the formation of General Motors 3—2

Germanium
- *See Transistors*

Gilliland, Harry J.
- Biography (P) 2—62
- Magnetic Test Accurately Compares Heat Extraction Properties of Quenching Media 2—31

Gordon, John F.
- Why a Technical Education? (Ed) (P) 1

Graham, Robert W.
- Biography (P) 2—63
- New Servo Controlled Simulator Measures Vehicle Ride and Roll 2—12

Green, Farno L.
- Biography (P) 2—63
- Studies of Radioisotopes Show Promise for Present and Future Applications in Engineering 2—17

Grotts, George W.
- Biography (P) 3—62
- Determine the Thermal Deflection of a Composite Beam (CP)
  - Problem 2—53
  - Solution 3—58

Guns
- Synchronous firing device eliminates gun mount rotation during firing of twin 40mm guns 2—26

Gyroscopes
- New gyroscope developed to aid in teaching of physics and allied sciences 4—21

H

Hafstad, Lawrence R.
- Biography (P) 1—59
- On Basic and Applied Science (Ed) (P) 2
- Research as Applied to Traffic Engineering 1—12

Hamilton, Allen B.
- Biography (P) 2—63
- Mechanical Vibration Analysis by Electrical Circuit Analogy 2—34
- The Use of an Electrical Model to Analyze Torsional Vibration in a Diesel Engine 4—2

Harris, Elwood K.
- Biography (P) 2—63
- Determine the Unbalanced Forces in a Converted V-8 Engine (CP)
  - Problem 1—56
  - Solution 2—54

Headlamps—Automotive
- Why cars have four headlights 1—19
- Headlamp lighting system for Firebird III turns on or off depending on ambient light intensity 4—33

Heat Exchangers
- The design and development of a blood heat exchanger for open heart surgery 3—38

Heat Transfer
- *See also Classroom Problems*
- Basic heat transfer fundamentals applied to heat exchanger used for open heart surgery 3—38

Heat Treatment
- Magnetic quench test aids in heat treatment of steels 2—31

Heckert, William F.
- A Discussion of Transistor Action and Manufacture 3—28
- Biography (P) 3—62

Heller, Werner H.
- Biography (P) 2—57
- Determine the Unbalanced Forces in a Converted V-8 Engine (CP)
  - Problem 1—56
  - Solution 2—54

Highway Safety
- Research can increase highway safety by improving the driver's competence 1—12
- Road coefficient of friction trailer aids in pinpointing highway surfaces worn smooth by tire polishing action 4—14

I

Industrial Wastes
- *See Waste Disposal*

Instrumentation
- *See also Control Systems*
- Find the resistance values of a thermocouple-potentiometer circuit (CP)
  - Problem 2—58
  - Solution 3—56

Internal Combustion Engines
- *See Engine Design*

Inventions
- *See Patents*

Isotopes
- *See Radioisotopes*

J

Jones, Fred E.
- Biography (P) 4—57
- The Filing and Maintenance of Patents in Foreign Countries 4—38

Joyce, Alex H.
- Biography (P) 4—58
- Dynamic Air Gaging Reduces Cycle Time in Precision Measurement 4—28

K

Kehrl, Howard H.
- Biography (P) 3—62
- Development of the Chevrolet W Engine: A New Concept in V-8 Engine Design 3—10

L

Lawless, Norman D.
- Biography (P) 1—59
- Manufacturing Engineers Develop New Method to Statically Balance Speedometer Drag Cups 1—36

Leach, Lisle
- Biography (inside back cover) 4

Lents, Ray
- Biography (inside back cover) 3

Lighting
- *See Headlamps—Automotive*

M

Macan, Thomas F.
- Biography (P) 3—60

Magnetism
- Fundamental knowledge of magnetism broadened by study of magnetic domain patterns 3—34

Magnus Effect
- *See Fluid Mechanics*

Manufacturing
- *See also New Equipment—Manufacturing, New Processes*
- Radioisotope measuring gage applied to the manufacture of storage battery separators 1—34
- Special balancing fixture aids in the manufacture of speedometer drag cups 1—36
- Special material handler aids in the manufacture of steering shaft couplings (inside back cover) 4

Mathematics
- *See also Classroom Problems*
- Development of an analog method for finding real and complex roots of higher order polynomial equations 1—29
- Mathematical analysis applied to programming of a digital computer 2—2

- The application of differential equations in developing electromechanical analogy concept for use in vibration analysis 2—34
- Trigonometric fundamentals applied to the development of a static balancing fixture 1—36

McLIMORE, HENRY S.
- Biography (P) 3—62
- Determine the Thermal Deflection of a Composite Beam (CP)
  - Problem 2—53
  - Solution 3—58

McPHERSON, DONALD H.
- Biography (P) 3—63
- Development of the Chevrolet W Engine: A New Concept in V-8 Engine Design 3—10

MEADE, KENNETH A.
- A Message to Educators About the GENERAL MOTORS ENGINEERING JOURNAL (P) 3—60

METHODS ENGINEERING
- Methods laboratory used to analyze pre-production operations (inside back cover) 3

MILITARY VEHICLES
- Electrodynamic vibrator applied to the development of military vehicles 1—2
- Synchronous firing device for twin 40mm guns eliminates gun mount rotation on armored mobile carriage 2—26

MITTELSTADT, ERIC
- Biography (P) 2—57
- Determine the Unbalanced Forces in a Converted V-8 Engine (CP)
  - Problem 1—56
  - Solution 2—54

N

NEW DEVELOPMENTS
- Blood heat exchanger developed for open heart surgery 3—38
- Electronic control system provides indoor proving grounds for transmissions 1—25
- GM engineers develop electromagnetic guidance system to steer vehicles automatically 2—60
- Magnus effect flow meter developed as a teaching device for fluid mechanics laboratory 4—12
- New gyroscope developed as an aid in teaching of physics and allied sciences 4—21
- Simple two wheel trailer measures road surface coefficient of friction 4—14
- Servo controlled simulator measures vehicle ride and roll 2—12
- Synchronous firing device developed for twin 40mm guns 2—26

NEW EQUIPMENT—MANUFACTURING
- Dynamic air gaging reduces cycle time in precision measurements 4—28
- Magnetic test accurately compares heat extraction properties of quenching media 2—31
- Manufacturing engineers develop new method to statically balance speedometer drag cups 1—36
- Radioisotope measuring gage applied to the production of storage battery separators 1—34

NEW PROCESSES
- Dielectric heating applied to body interior trim fabrication 4—7
- Radioisotope wear study techniques applied to evaluate the lubricity of oils 2—21

NOON, WALTER D.
- Biography (P) 1—59
- Determine the Operating Characteristics of a Drive and Control System for a Transfer Mechanism (CP)
  - Problem (Vol. 4) 4—49
  - Solution 1—48

O

O'KANE, G. J.
- Biography (P) 1—59
- Planning and Operating an Industrial Waste Disposal Plant for a New Plating Facility 1—8

ONKSEN, GEORGE W.
- Biography (P) 1—59
- Why Cars Have Four Headlights 1—19

OPEN HEART SURGERY
- *See Heat Exchangers*

ORSAT ANALYSIS
- *See Classroom Problems*

OSBORN, C. R.
- On the Challenge of the Space Age (Ed) (P) 4

P

PATENTS
- JOURNAL pages include a general discussion of patent law followed by descriptions of individual patents granted to GM personnel 1—40, 2—42, 3—41, 4—38

PATENT SECTION—GENERAL MOTORS
- A discussion of the organization and function of the GM Patent Section 2—42

PATENT EXHIBIT
- Progress under U.S. patent system told in public exhibit 1—47

PHYSICS
- *See also Gyroscopes*
- Fundamental knowledge of magnetism broadened by study of magnetic domain patterns 3—34

PLANT DESIGN
- Planning and operating an industrial waste disposal plant for a new plating facility 1—8

PRIOR, JOHN R.
- Applying the Electrodynamic Vibrator to the Development of Military Vehicles 1—2
- Biography (P) 1—60

PRODUCT DESIGN
- *See also Engine Design, Classroom Problems*
- A discussion of some of the considerations necessary in the design and application of transistors 3—28
- Automobile radios are designed to compensate for specific environmental conditions 2—38
- Digital computer aids in the design of air conditioning system components 3—17
- Dual headlighting system provides improved optical characteristics 1—19
- Heat exchanger designed for use in open heart surgery 3—38
- Stress analysis techniques applied to improve casting designs 4—22
- The design and development of a synchronous firing device for twin 40mm guns 2—26

PRODUCTION PLANNING
- Cycle time in precision measurement reduced by dynamic air gaging 4—28
- New balancing fixture speeds the static balancing of speedometer drag cups 1—36

PUBLIC RELATIONS
- On Science, Engineering, and Public Relations (Ed) by Anthony DeLorenzo (P) 3

Q

QUALITY CONTROL
- Radioisotope measuring gage aids in quality control of storage battery separators 1—34

R

RADIOISOTOPES
- Radioisotope measuring gage applied to the production of storage battery separators 1—34
- Radioisotope studies show promise for present and future applications in engineering 2—17
- Radioisotopes used to evaluate lubricity of oils used in refrigerant compressor 2—21

RADIOS
- *See Automobile Radio Design*

RAUSCH, JOHN T.
- Biography (P) 3—63
- Development of the Chevrolet W Engine: A New Concept in V-8 Engine Design 3—10

RAVITCH, LOUIS H.
- Biography (P) 4—58
- Some Applications of Stress Analysis Techniques in Improving Casting Designs 4—22

REISING, PAUL J.
- Biography (P) 3—63
- The Patentability of New Uses for Old Materials and Devices 3—41

RENIUS, RICHARD P.
- Biography (P) 4—57

RESEARCH
- On Basic and Applied Science (Ed) by Lawrence R. Hafstad 2
- Research as applied to traffic engineering 1—12
- Single crystal iron whiskers adapted to the basic study of magnetic domains 3—34

(CP) Classroom problem; (Ed) Editorial—always appears on unnumbered inside front cover; (P) Portrait

RITTMAN, ALBERT D.
A Discussion of Transistor Action and Manufacture - - - - - - 3—28
Biography (P) - - - - - - - - 3—63

RUFF, KENNETH E.
Biography (P) - - - - - - - - 4—58
Dynamic Air Gaging Reduces Cycle Time in Precision Measurements - - - - - - - - - 4—28

S

SACCA, DEMETRIO B.
Biography (P) - - - - - - - - 3—63
The Design and Development of a Blood Heat Exchanger for Open Heart Surgery - - - - - - - 3—38

SAFETY
See Highway Safety

SCANES, ERNEST W.
Biography (P) - - - - - - - - 4—57

SCHRENK, GEORGE L.
An Analog Method for Finding the Real and Complex Roots of Higher Order Polynomial Equations - - - - - - - - - 1—29
Biography (P) - - - - - - - - 1—60

SCHWARZ, BERTRAM A.
A Discussion of Some Factors Affecting Reproduced Radio Sounds in an Automobile - - 2—38
Biography (P) - - - - - - - - 2—64

SCIENCE
On Basic and Applied Science (Ed) by Lawrence R. Hafstad - 2
On Science, Engineering, and Public Relations (Ed) by Anthony DeLorenzo - - - - 3

SCOTT, GIFFORD G.
Biography (P) - - - - - - - - 3—64
Single Crystal Iron Whiskers Adapted to the Basic Study of Magnetic Domains - - - - - 3—34

SEMICONDUCTORS
See Transistors

SERVOMECHANISMS
See also Control Systems
Electrohydraulic servomechanisms developed for accurate control applications (inside back cover) 2
Servo controlled, electrohydraulic simulator used to test vehicle ride and roll - - - - - - - - 2—12

SHAFER, RONALD E.
Biography (P) - - - - - - - - 2—64
New Servo Controlled Simulator Measures Vehicle Ride and Roll 2—12

SKEELS, PAUL C.
Biography (P) - - - - - - - - 4—58
Simple Two-Wheel Trailer Measures Road Surface Coefficient of Friction - - - - - - - - - 4—14

SPEAKING ENGAGEMENTS—GM ENGINEERS
See Engineers

STALLARD, RALPH M.
Biography (P) - - - - - - - - 4—58
The Application of Dielectric Heat to Body Interior Trim Fabrication - - - - - - - - 4—7

STEERING EQUIPMENT—AUTOMOBILE
Firebird III, experimental gas turbine powered car, has automatic steering system - - - - 4—33
GM engineers develop electromagnetic guidance system to steer vehicles automatically - - 2—60

STRESS ANALYSIS
See also Classroom Problems
Stress analysis techniques applied to improve casting designs - - 4—22

STRICKLAND, GEORGE H.
Biography (P) - - - - - - - - 2—64
Organization and Function of the GM Patent Section - - - - - 2—42

SUSPENSION SYSTEMS—AUTOMOTIVE
Determine the effect of changes in ambient temperature on an automotive air suspension system (CP)
Problem - - - - - - - - - 3—55
Solution - - - - - - - - - 4—54
Servo-controlled, electrohydraulic simulator used to test vehicle suspension systems - - - - - 2—12

SWANSON, RONALD J.
Biography (P) - - - - - - - - 1—55
Determine the Center of Gravity Location of a Passenger Car (CP)
Problem - - - - - (Vol. 4) 4—45
Solution - - - - - - - - - 1—54

T

TABB, JOHN B.
Biography (P) - - - - - - - - 4—57

TESTING
Dynamometer control system simulates actual proving ground tests on transmissions in the laboratory - - - - - - - - - 1—25
Electrodynamic vibrator applied to simulation testing in the development of military vehicles - - - - - - - - - - - 1—2
Magnetic test accurately compares heat extraction properties of quenching media - - - - - 2—31
Radioactive wear study techniques used in test to evaluate lubricity of oils - - - - - - - 2—21
Servo-controlled, electrohydraulic simulator tests vehicle suspension systems - - - - - - - - 2—12

TORBIT, RAY
Biography (P) - - - - - - - - 2—57
Determine the Unbalanced Forces in a Converted V-8 Engine (CP)
Problem - - - - - - - - - 1—56
Solution - - - - - - - - - 2—54

TRAFFIC ENGINEERING
Research as applied to traffic engineering - - - - - - - - 1—12

TRANSISTORS
A discussion of the principles and action of transistors and some aspects of their manufacture - 3—28

TRATHEN, WESLEY J.
Biography (P) - - - - - - - - 4—59
The Application of a Magnus Effect Flow Meter as a Laboratory Teaching Device - - - - 4—12

TUCK, HARVEY R.
Biography (P) - - - - - - - - 3—64
The Application of a Digital Computer to the Design of Air Conditioning System Components - - - - - - - - - - - - 3—17

V

VIBRATION ANALYSIS
A review of the theory behind electro-mechanical analogy as applied to vibration analysis - - 2—34
Electrodynamic vibrator aids in vibration analysis of military tank components - - - - - - 1—2
Electro-mechanical analogy provides simple approach to vibration analysis of Diesel engines - 4—2

VICKERS, PAUL T.
Biography (P) - - - - - - - - 2—64
Testing Paper Gas Turbines: Applying the Digital Computer to Engine Design - - - - - - 2—6

W

WANTTAJA, GLENN
Biography - (inside back cover) 2

WASTE DISPOSAL
Planning and operating an industrial waste disposal plant for a new plating facility - - - - - 1—8

WEST, PHILIP
Biography (P) - - - - - - - - 1—60
Determine the Operating Characteristics of a Drive and Control System for a Transfer Mechanism (CP)
Problem - - - - - (Vol. 4) 4—49
Solution - - - - - - - - - 1—48

WHITAKER, R. O.
An Analog Method for Finding the Real and Complex Roots of Higher Order Polynomial Equations - - - - - - - - - 1—29
Biography (P) - - - - - - - - 1—60

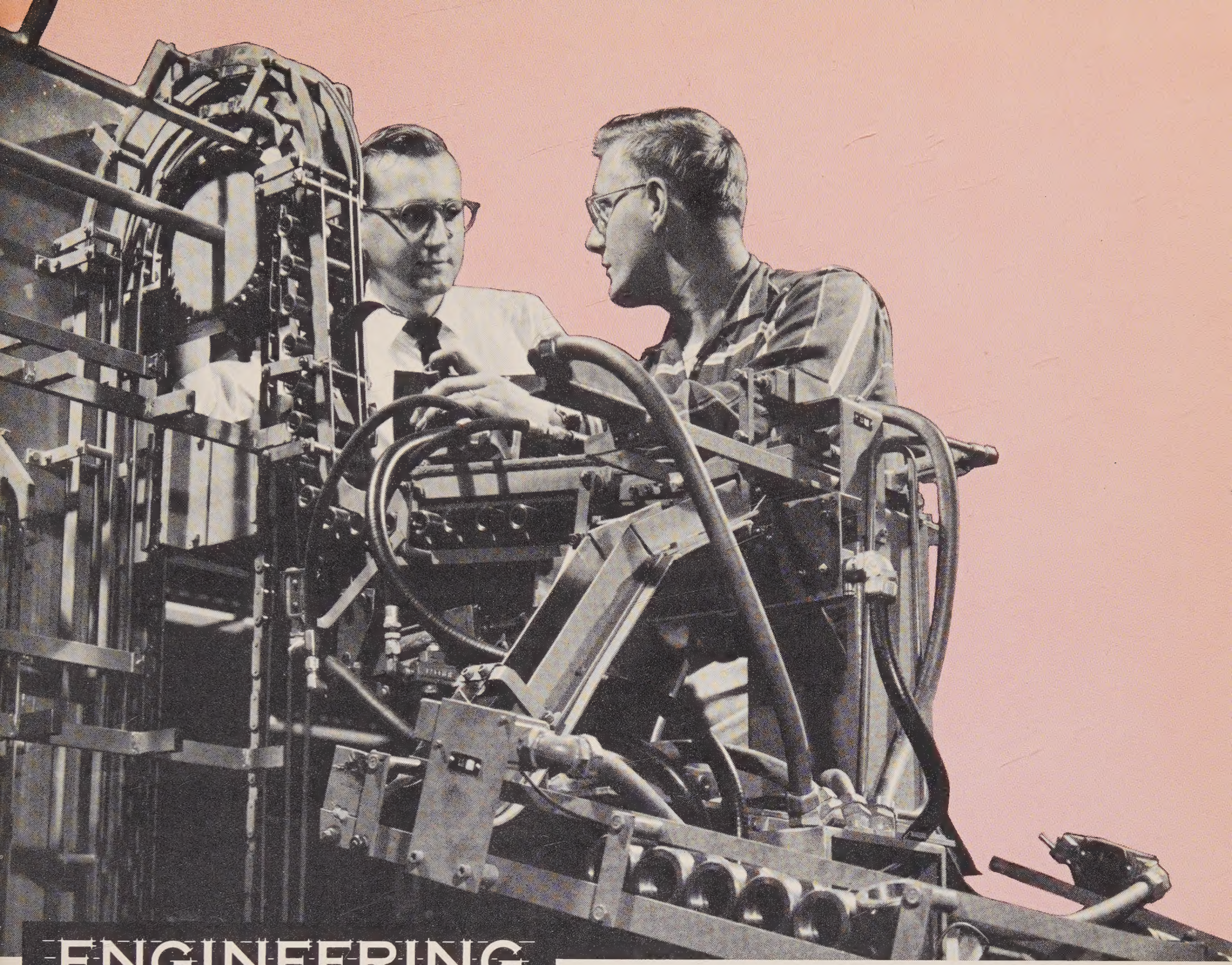
WOCHOLSKI, LEONARD B.
Biography (P) - - - - - - - - 4—59
Determine the Effect of Changes in Ambient Temperature on an Automotive Air Suspension System (CP)
Problem - - - - - - - - - 3—55
Solution - - - - - - - - - 4—54

WOODFIELD, KENNETH W.
Biography (P) - - - - - - - - 4—59
Determine the Static and Total Temperatures of a High Temperature, High Velocity Gas Stream (CP)
Problem - - - - - - - - - 3—53
Solution - - - - - - - - - 4—47

WOODS, RONALD O.
Biography (P) - - - - - - - - 3—60

Z

ZIMMERMAN, PATON M.
Biography (P) - - - - - - - - 4—59
The Application of Dielectric Heat to Body Interior Trim Fabrication - - - - - - - - 4—7



ENGINEERING

ASSIGNMENT IN GM

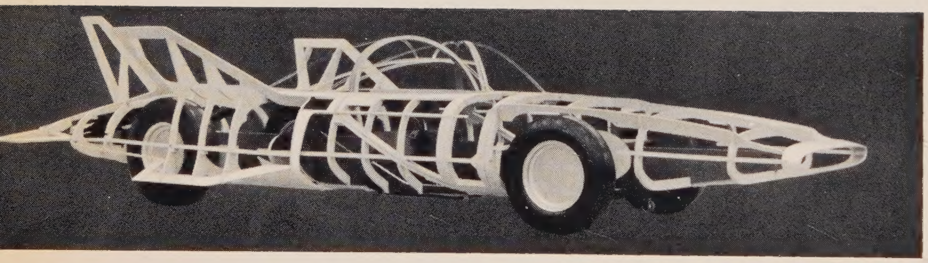
Pioneering in the field of manufacturing offers unusual opportunities for the versatile young engineer who can work with his hands as well as his head. The designing, installing and "debugging" of special machinery, in particular, abounds in such opportunities.

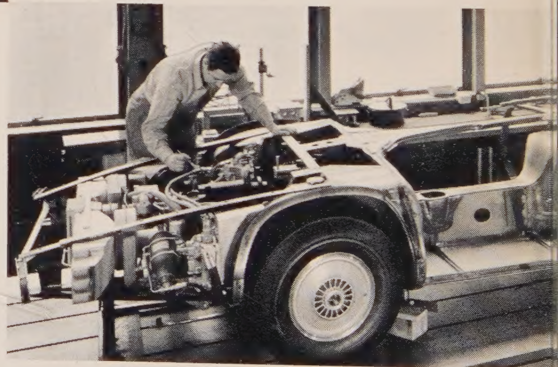
One example of a special machine in General Motors is a material handler (above) which is part of a continuous flow line for manufacturing steering shaft couplings at Saginaw Steering Gear Division, Saginaw, Michigan. This machine removes the couplings as they pass through a centerless grinder and holds them in an oriented, ready position. It then places them as needed, six at a time, in a milling machine for the next operation. Equipment of this type is usually designed and built especially for the specific application, and final "debugging" must be done after the machine is installed rather than on the drawing board.

Eugene Ezop, left, is a 1957 General Motors Institute graduate in industrial engineering. Throughout his college course, he took as many optional studies as possible in the field of electronics. He now is employed as a college graduate-in-training in the Process Engineering Department at Saginaw Steering Gear Division. Most of his duties have to do with development of induction heating equipment; however, as in this case, the ability to handle different assignments is important.

Conferring with Mr. Ezop is Lisle Leach, an engineering technician in the Saginaw Steering Gear Process Development Laboratory. These two young men worked as a team in helping to ready this complicated piece of equipment for duty on the production line.

GENERAL MOTORS ENGINEERING JOURNA





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